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A study of film type condensation inside horizontal tubes

Sullivan, Thomas J., Jr.; Leuschner, Robert J.

Cambridge, Massachusetts; Massachusetts Institute of Technology

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A STUDY OF FILM TYPE CONDENSATION INSIDE HORIZONTAL TUBES

THOMAS J. SULLIVAN, JR.
ROBERT J. LEUSCHNER

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A STUDY OF FILM TYPE CONDENSATION INSIDE HORIZONTAL TUBES

by

Thomas J. Sullivan, Jr., Lieutenant, U. S. Navy
B. S. in M. E., Carnegie Institute of Technology, 1942

Robert J. Leuschner, Lieutenant (j.g.), U. S. Navy
B. S., U. S. Naval Academy, 1945

Submitted by the Authors in Partial Fulfillment of the Requirements for the Degree of

NAVAL ENGINEER

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

1951

Thesis
S562

ABSTRACT

"A STUDY OF FILM TYPE CONDENSATION
INSIDE HORIZONTAL TUBES"

Names of Authors: Thomas J. Sullivan, Jr.
Robert J. Leuschner

Submitted for the Degree of Naval Engineer in the Department of Naval Architecture and Marine Engineering on May 18, 1951

This thesis reports the results of temperature measurements made by thermocouples installed on the exterior surface of a single, horizontal copper tube while saturated steam vapor was condensing within. Cooling was effected by cross flow of cooling water with one pass. It describes an experimental technique for making temperature measurements. The effect of vapor velocity on distorting the build up of the liquid film is shown. There is included a theoretical method for the prediction of local coefficients of heat transfer developed using the assumptions of the Nusselt theory. Overall coefficients of heat transfer were determined at two different steam conditions.

15652

ACKNOWLEDGEMENT

The authors wish to express their appreciation to Professor A. L. Hesselschwerdt, Jr., their thesis supervisor, for his assistance and constructive criticism in outlining and carrying through the research work for this thesis; to Professor W. M. Rohsenow for his guidance in resolving the problems of heat transfer encountered in the pursuit of this investigation; to Professor L. R. Vianey for his aid in the constructional aspects of the equipment used and in solving the practical problems of thermal measurements; and to other members of the staff and faculty for their wholehearted cooperation.

Cambridge, Massachusetts
May 18, 1951

Professor J. S. Newell
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge, Massachusetts

Dear Sir:

In accordance with the requirements for the degree of Naval Engineer, we submit herewith a thesis entitled, "A Study of Film Type Condensation Inside Horizontal Tubes."

Respectfully,

TABLE OF CONTENTS

SUMMARY

The object of this thesis investigation was to study film type condensation in horizontal tubes; to attempt to find a theoretical method for the prediction of local and mean heat transfer coefficients for the water film which forms inside the tubes; and to prove the theory by the results of experimental work. These objectives were not fully realized. It is hoped that later investigators might find here information that will aid them in avoiding pitfalls which were encountered during the course of this work.

Why should one be interested in the study of film type condensation since, by the use of "promoters" such as benzyl mercaptan, dropwise condensation can provide greater rates of heat transfer?

Film type condensation is encountered in refrigeration cycles. Unlike power cycles, refrigerant vapors are condensed at relatively high pressures. It becomes advisable for the vapor to be introduced inside the tubes of the condenser and not into the shell. Design of these condensers would be greatly facilitated by the ability to predict accurately the heat transfer coefficients encountered in practice.

In this work steam vapor was selected for the condensing medium. The properties of steam are much more accurately delineated than for refrigerant vapors. If a method and apparatus could be developed to determine heat transfer coefficients for steam vapor films, it might be easily adapted for use with refrigerant vapors.

The method for the experimental work was to introduce saturated steam vapor into a six foot length of copper tubing. Cooling was effected by circulating water from city mains passed at right angles to the axis of the tube. Temperature measurements were made by installing thermocouples in a line on the outer wall of the tube. The test section of copper tubing was so fitted that it could be rotated on the axis of the tubing. This feature permitted recording of tube wall temperatures at any angle around the circumference and along the length of the tube.

By means of such apparatus, a measured heat transfer was allowed to take place between two points of known temperature difference, the condensing vapor inside the tube and the outer surface of the tube wall. Two thermal resistances lay in the path of heat flow, the resistance of the water film forming inside the tube and the resistance of the copper tube wall. The latter resistance is nearly negligible compared to the former. Both can be evaluated readily. After the apparatus had been operated several important results were determined from this, the initial one of a series of studies on film type condensation.

The test section had been made of great length to minimize heat loss at ends of the tube. With the high rates of heat transfer encountered, this caused excessive velocity of the steam entering the tube. This high velocity wipes off the water film from the inner wall, increases heat transfer rates at the entrance end of the tube, and does not allow valid proof of a Nusselt type theory.

The experimental technique of measuring tube wall temperatures by means of thermocouple junctions with leads immersed in cooling water is shown to be workable and valuable. The results indicate detectable temperature differences from point to point, both circumferentially and along the length, on the tube surface.

Avoidance of variable cooling water temperature, as would be encountered by use of a counterflow heat exchanger, is demonstrated in the equipment used. A constant wall temperature is an essential assumption to theory.

A mean heat transfer coefficient for steam vapor was found to be about 2000 Btu/hr.-ft.^{2}–°F.

It is recommended that for future observers that the length of the condensing section be shortened relative to the tube diameter. This will reduce the effect of velocity at the entrance section, will provide a horizontal tube with greater resistance to deflection, and facilitate horizontal alignment. The heat transfer rate should be reduced by reducing the temperature difference between the cooling water and the condensing vapor and not by reducing the quantity of cooling water. Greater attention must be devoted to keeping the condensing surface clean and free of dirt deposits. The small heating boiler used for the experimental work of this thesis is considered to be not suited to scientific investigation.

INTRODUCTION

When a pure saturated vapor comes into contact with a colder surface there is a transfer of heat which takes place between the surface and the vapor. Some of the vapor condenses to form liquid on the surface. If the surface is vertical, for example, the liquid will flow under the action of gravity down the wall accumulating more and more liquid as it moves. The liquid "wets" all areas of the surface forming a film which is resistant to the transfer of heat between the vapor and the wall. This is called film type condensation.

In 1916, Nusselt derived theoretical relations for the prediction of the thickness of this film, and hence the local coefficient of heat transfer for any point on a vertical wall. By integration a mean coefficient of heat transfer may be found for any particular dimensions of wall.

Nusselt's derivation includes important and limiting assumptions:

(a) that the flow of the liquid film is due to the force of gravity alone.

(b) that the flow of the film is laminar; has a parabolic velocity distribution across any thickness; and that the velocity at the wall is zero.

(c) that the temperature difference between the vapor and the wall surface is constant at all points.

The first two assumptions place a restriction on the use of the theory. If the vapor is flowing with an appreciable velocity, the drag which it exerts at the liquid-vapor interface is a force unaccounted for by the theory. The effect of vapor velocity is to disturb the formation of a laminar film. The third assumption is difficult to find in nature. Since the flow of heat from the condensing vapor must go to some cooling medium, that medium increases in temperature. Then the temperature difference between the condensing vapor and the cooling surface is different at one point than it is at another.

There have been many investigators who have turned their attention to the problem of predicting heat transfer coefficients for vapors condensing on the outside of tubes. This activity is a natural consequence of the wide use of surface condensers in the fields of power generation and chemical engineering. With power cycles, for example, the condensing vapor has a large specific volume and a low pressure after it has passed through the expansion process. It is desirable to introduce this large volume of vapor into the shell of the condenser. The cooling water is passed through tubes; the vapor condenses on the outside of the tubes.

A refrigeration cycle is a reversed power cycle. The working fluid leaves the compression process at a relatively high pressure and low specific volume. It would be uneconomical to construct a heavy condenser shell capable of withstanding the pressure. The working fluid of this type of

cycle is usually introduced into small tubes and the cooling water is passed around the outside of these tubes in order to condense the refrigerant vapor.

Investigation of film type condensation inside horizontal tubes is almost nil. The sole known investigator in this field is Max Jakob. (Reference 2). His equipment consisted of a short section of tubing, cooled on the outside and into which steam vapor was introduced. A probe carrying three thermocouples was fitted so that it might be used to explore the inside of the tube. The probe could enter the tube or be withdrawn. It could be rotated. The thermocouples were held tight against the interior surface of the tube wall so that the temperature of any point on the interior surface could be measured. Jakob reports detectable temperature differences around the circumference of the tube by this method.

It seems that insertion of thermocouple leads into the liquid film would disturb the natural build up of that film; therefore, observed readings of the wall temperature would disagree with those of an undisturbed film. Also, since the thermocouple leads would be surrounded by high temperature steam vapor a correction would have to be made for the conduction of heat in the leads down to the junction point.

The problem of the prediction of heat transfer coefficients in horizontal tubes seems to consist of two parts. First, to provide conditions within the experimental apparatus which will approximate the assumptions of the Nusselt theory

in order to prove or to disprove the applicability of that theory. Secondly, to provide conditions within the experimental apparatus which approximate conditions found in commercial condensers. An orderly, scientific study of the problem requires that each part of it be done separately.

In the initial study of the problem then, one would desire to have negligible velocity of steam vapor across the liquid-vapor interface and he would desire to have a condensing surface temperature which was essentially constant over all points. In the second part of the problem, one would allow vapor velocities to effect whatever distortion of the liquid film that they will and would allow the condensing surface temperature to vary from point to point in the natural manner of a counter-flow or cross-flow condenser.

PROCEDURE

After several consultations with Professors Hesselschwerdt and Rohsenow, the basic elements of the design were agreed upon. The test section was to six feet in length, made of copper, and 3/4" ID. The test section as actually used was a 3/4" ID type "L" copper tube. If time permitted, runs were also to be made with a two foot and a four foot section.

Originally the water plenums were to be 6" pipe with 12 diametrically opposite holes on each side of the plenums. The apparatus employs 5" plenums since they were obtained gratis and believed to do the job equally as well as the 6" pipes.

The plenums were connected by rubber tubing instead of valves as was originally intended in order to conserve limited funds.

The copper test section was grooved at intervals of 6" (twelve grooves in all) and a #28 copper-constantan thermocouple soft soldered in each groove. The thermocouples were along a straight line on the tube and, after being soldered in place, were wrapped around the tube once in order to minimize the conduction losses. The thermocouple wires were then thrice coated with polystyrene, an electrical insulator, to avoid any short circuits. The thermocouples were then gathered consecutively at each station and lead to the discharge end of the test section water plenum. This was done to avoid their being unduly swashed back and forth in the pipe with the possibility of rubbing off the protective insulation.

The thermocouples were calibrated in accordance with standard practice, and an enlarged graph of millivolts versus temperature was drawn to insure the best possible results for the readings obtained.

The test tube was checked for horizontal alignment by use of a water level consisting of a piece of garden hose with a section of glass tubing in it. The actual alignment being done by four set screws in the base of the structure holding the apparatus.

Manometers were inserted at both ends of the test section as were thermocouples.

After the apparatus was completed a dry run was made to ascertain the accuracy of the thermocouples, and they were all found to be operating satisfactorily.

The procedure followed to obtain the data is a step-by-step process of starting up the apparatus coupled with the ordinary routine of taking thermocouple readings.

1. The copper test section is oriented as to position desired for the run, and the needle valve controlling the flow of condensate is opened.

2. The boiler is lighted off and brought up to the pressure desired. Pressure on the boiler is controlled by moving the sliding weight on the balancing arm.

3. After the boiler is up to pressure and the air is ejected from the system, the needle valve is closed and the cooling water to the test section is turned on and the condensate is thus allowed to fill up the tank containing the gage glass.

4. The cooling water to sub-cool the condensate is turned on.

5. The pet cock on the discharge end of the copper test section is then cracked to allow a steady stream of air and steam to flow out.

6. When the water in the small tank is at gage glass level the needle valve is adjusted to obtain a steady flow of condensate and hence, equilibrium.

7. Once equilibrium is obtained, the condensate is weighed and the thermocouples read.

Having obtained a set of readings for one position of the copper test section, it is possible to rotate the test section without disturbing any of the valves on the apparatus. This is the case since low pressure steam is being used. It is recommended that if higher pressures are used with this apparatus that this procedure be abandoned. The rotation is brought about by slightly breaking the unions at either end of the test section and then rotating the tube. This procedure allows equilibrium to be remaintained in a minimum of time.

The condensate is weighed for a fifteen (15) minute period to obtain the Q total which is used in the formula

$$Q = U A \Delta t_m$$

Q then being the product of $w_{cond} \times h_{fg}$

A, the area of the test section, is known from the dimensions.

Δt_m is found by an unusual approach. The process is to average the readings for each orientation of the pipe for each thermocouple, giving an average temperature for each thermo-

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[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

couple. Then the twelve thermocouple readings are averaged to give the mean temperature for each orientation of the pipe. The four mean temperatures are then averaged to give the final average Δt_{mean} (that is top, two sides, and bottom). It being assumed in this process that each reading was practically constant over a quadrant of the test section. The temperature of the steam being known, it is then a simple matter to obtain the Δt_{mean}. U is then simply a matter of running through the formula above.

In the computation of the mean temperature for the low pressure run, it can be seen that station 4 is far out of line. This was due to a broken glass tube in the reference junction, and the reasonable point between stations 3 and 5 was used in the computations, as that of station 4.

In determining the quantity Q total, the weight of condensate times h_{fg} was used. This is not entirely correct since the steam was not dry saturated, however, it was so close to being dry saturated that it was deemed unnecessary to make the slight change in h_{fg} since it would not have materially affected the answers obtained.

[illegible]

[illegible]

[illegible]

RESULTS

First Run: Pressure 20.65 psia. Saturation Temperature 229.4°F.

Average temperature along test section

Thermocouple Junction	Top of Tube	Side of Tube	Bottom of Tube
1	217.4	212.9	210.9
2	206.6	204.4	203.6
3	205.3	200.4	199.7
4	202.4	199.4	197.4
5	194.5	193.3	191.7
6	192.8	189.9	186.5
7	195.7	190.6	180.8
8	192.0	188.8	180.7
9	186.4	183.8	174.7
10	192.4	187.7	174.5
11	187.0	186.5	164.4
12	192.0	191.5	164.8
Average temperature	197.1	194.1	185.8
Mean wall temperature	192.8		

$\Delta t = 36.6°F$

$U_{overall} = 2110$ Btu/hr-ft^2-°F

$h_{vapor\ side} = 2180$ Btu/hr-ft^2-°F

Pressure drop in pipe not recorded in this run.

Average weight of condensate 111 lbs/hr.

Second Run: Average Pressure 17.15 psia. Saturation Temperature 219.0°F.

Average Temperature along test section

Thermocouple Junction	Top of Tube	Side of Tube	Bottom of Tube
1	201.6	201.7	203.8
2	194.2	191.4	194.0
3	190.2	186.8	192.0
4	175.7	172.7	172.9
5	184.5	180.2	179.9
6	180.4	178.5	174.5
7	185.3	181.7	172.5
8	192.5	193.9	170.6
9	177.0	181.3	161.6
10	174.0	180.8	151.3
11	168.0*	167.8	144.7
12	160.0*	160.0*	140.0*

* Values are approximations as to the true temperature at the points so marked. These approximations were made because the readings taken were definitely too low. The system was beginning to get air bound, and the air binding gave rise to such low temperature readings.

Average temperature	182.6	181.4	172.3

Mean wall temperature 179.4°F

Δt = 39.6°F

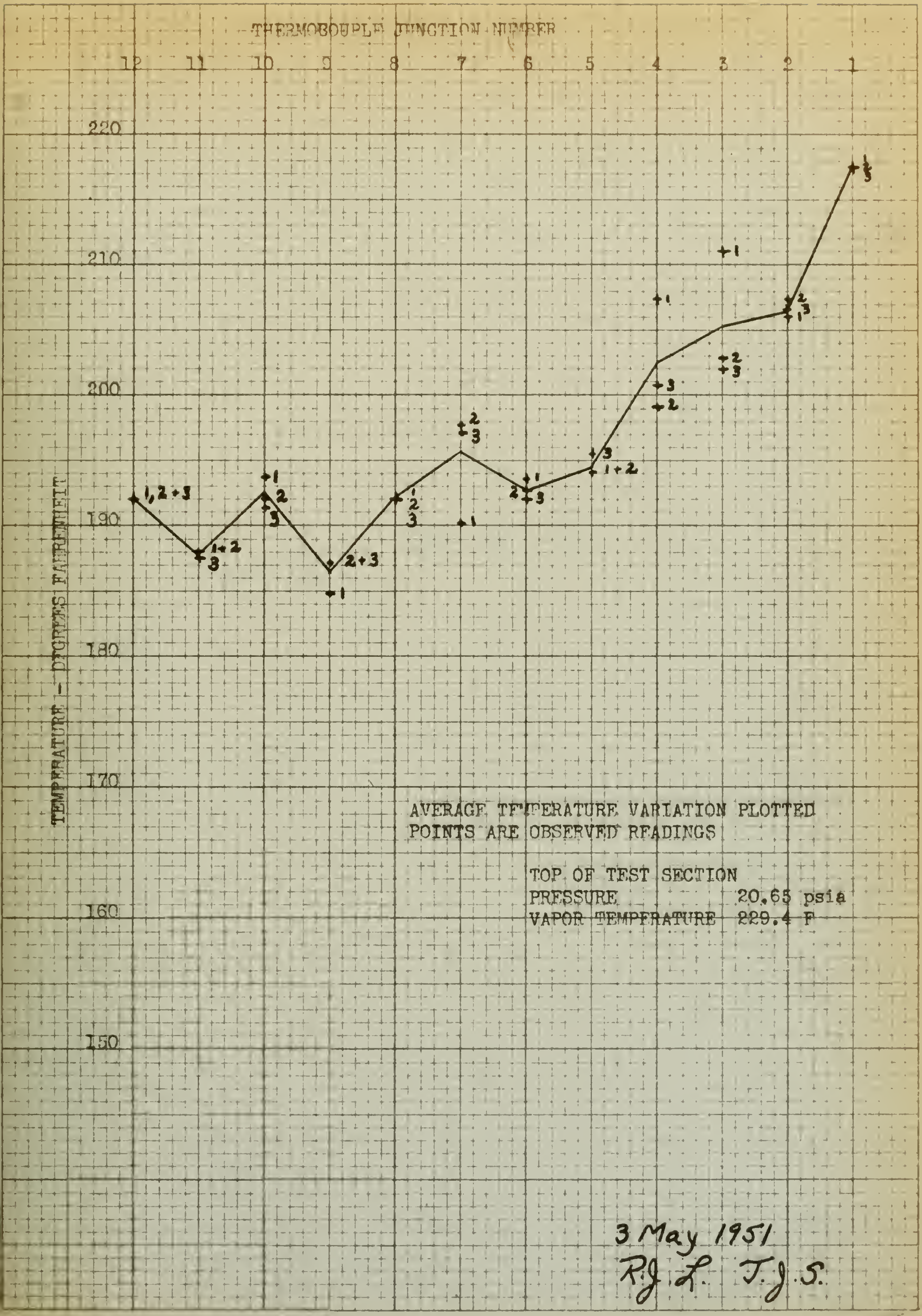
$U_{overall}$ = 1780 Btu/hr-ft^2-°F

$h_{vapor\ side}$ = 1840 Btu/hr-ft^2-°F

Average weight of condensate 93 lbs/hr.

Mean measured pressure difference 0.753" of Hg.

FIGURE I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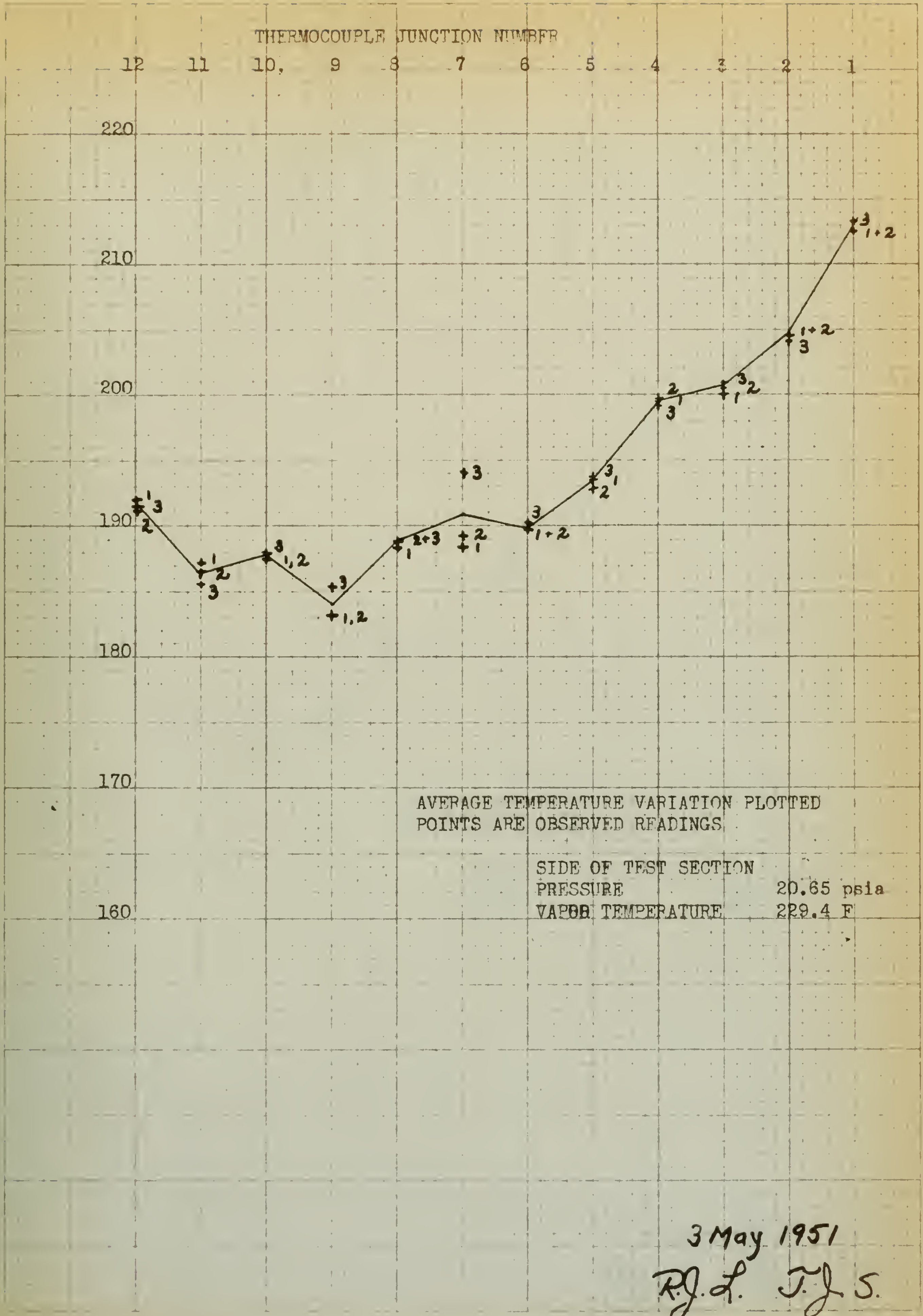
200
190
180
170
160
150
AVERAGE TEMPERATURE VARIATION PLOTTED
POINTS ARE OBSERVED READINGS
TOP OF TEST SECTION
PRESSURE 20.65 psia
VAPOR TEMPERATURE 229.4 F
3 May 1951
R.J.L. T.J.S.

FIGURE II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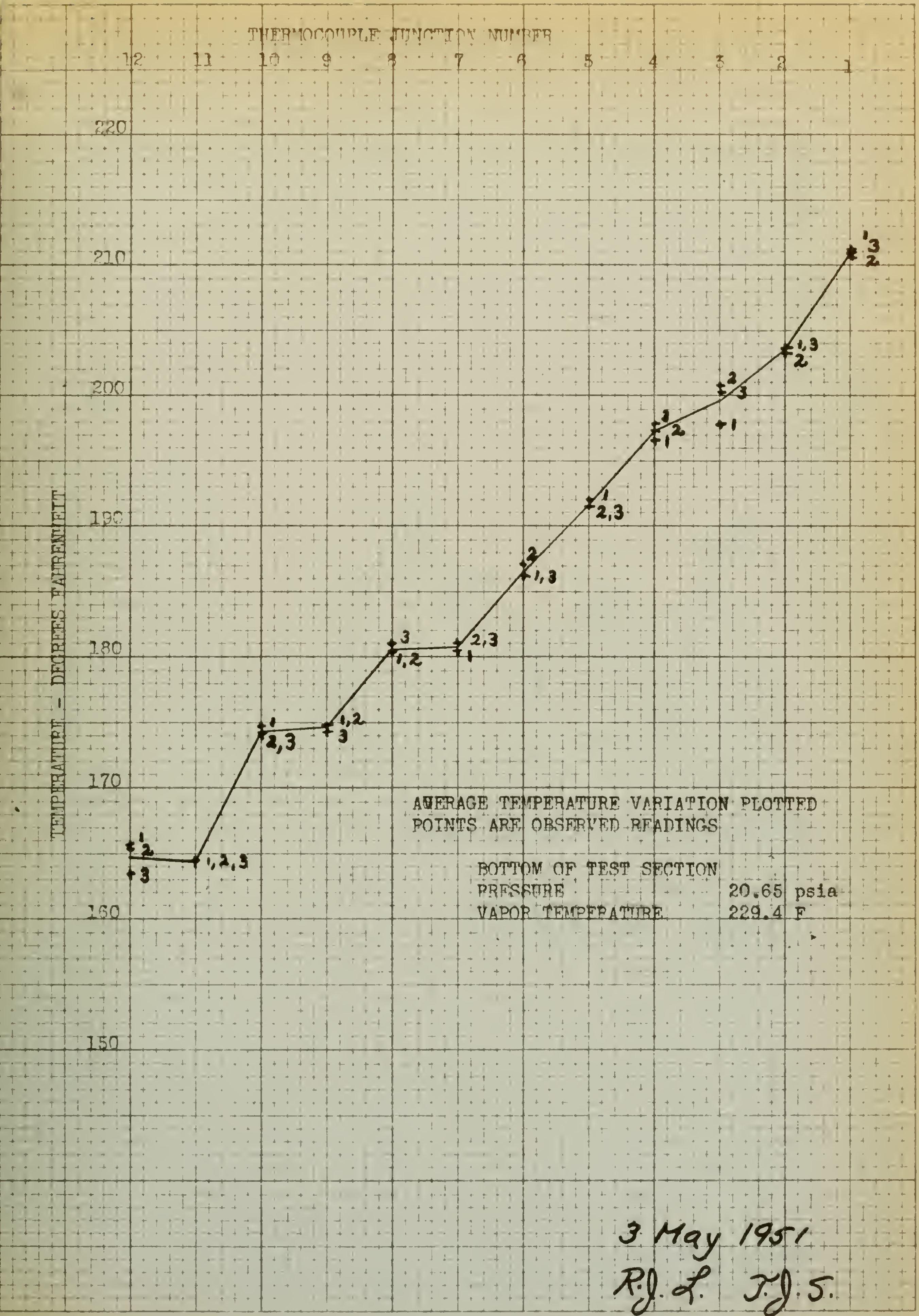
AVERAGE TEMPERATURE VARIATION PLOTTED
POINTS ARE OBSERVED READINGS
SIDE OF TEST SECTION
PRESSURE 20.65 psia
VAPOR TEMPERATURE 229.4 F
3 May 1951
R.J.L. T.J.S.

FIGURE III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220
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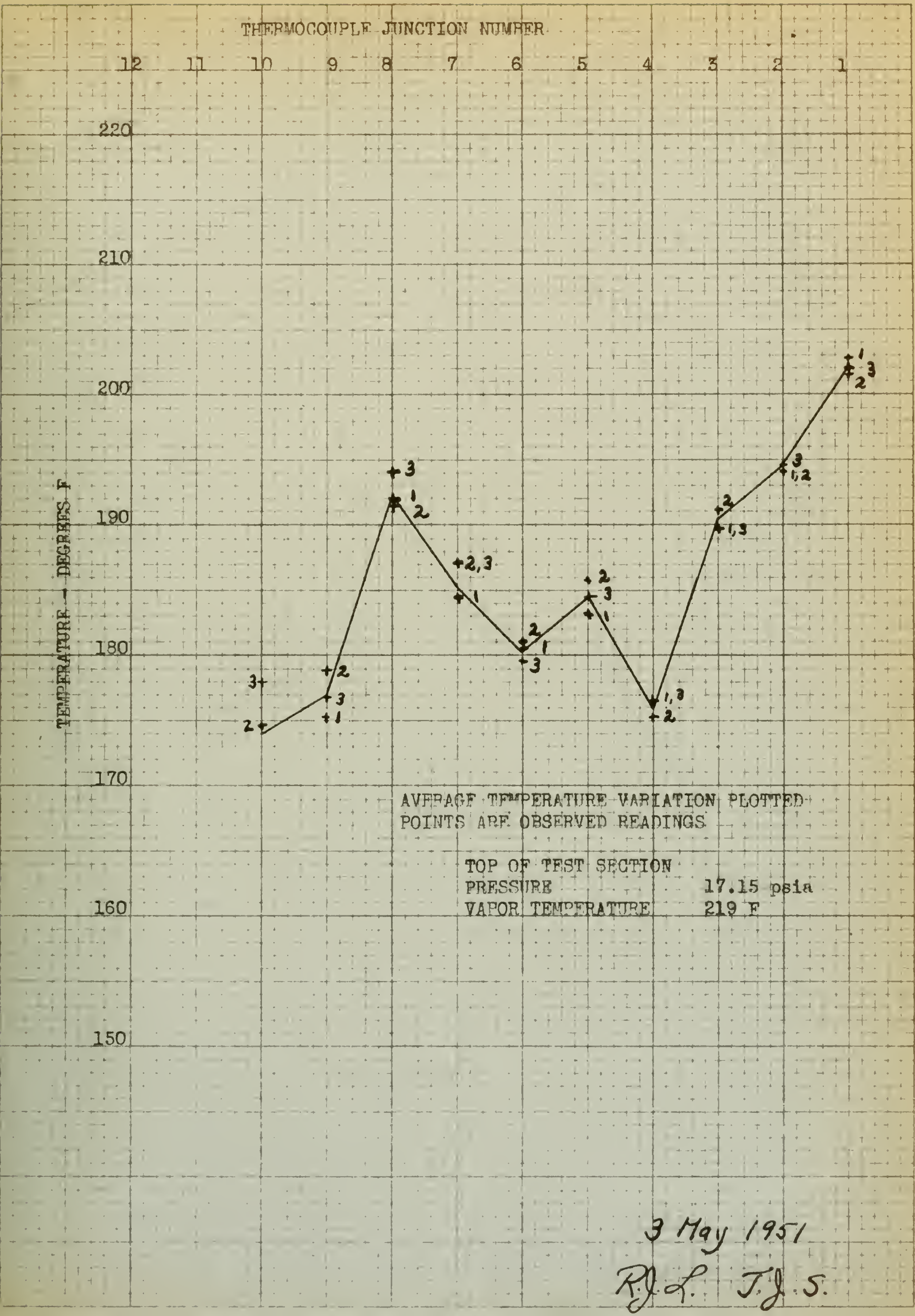
200
190
180
170
160
150
TEMPERATURE - DEGREES FAHRENHEIT
AVERAGE TEMPERATURE VARIATION PLOTTED
POINTS ARE OBSERVED READINGS
BOTTOM OF TEST SECTION
PRESSURE 20.65 psia
VAPOR TEMPERATURE 229.4 F
3 May 1951
R.J. L. T.J. S.

FIGURE IV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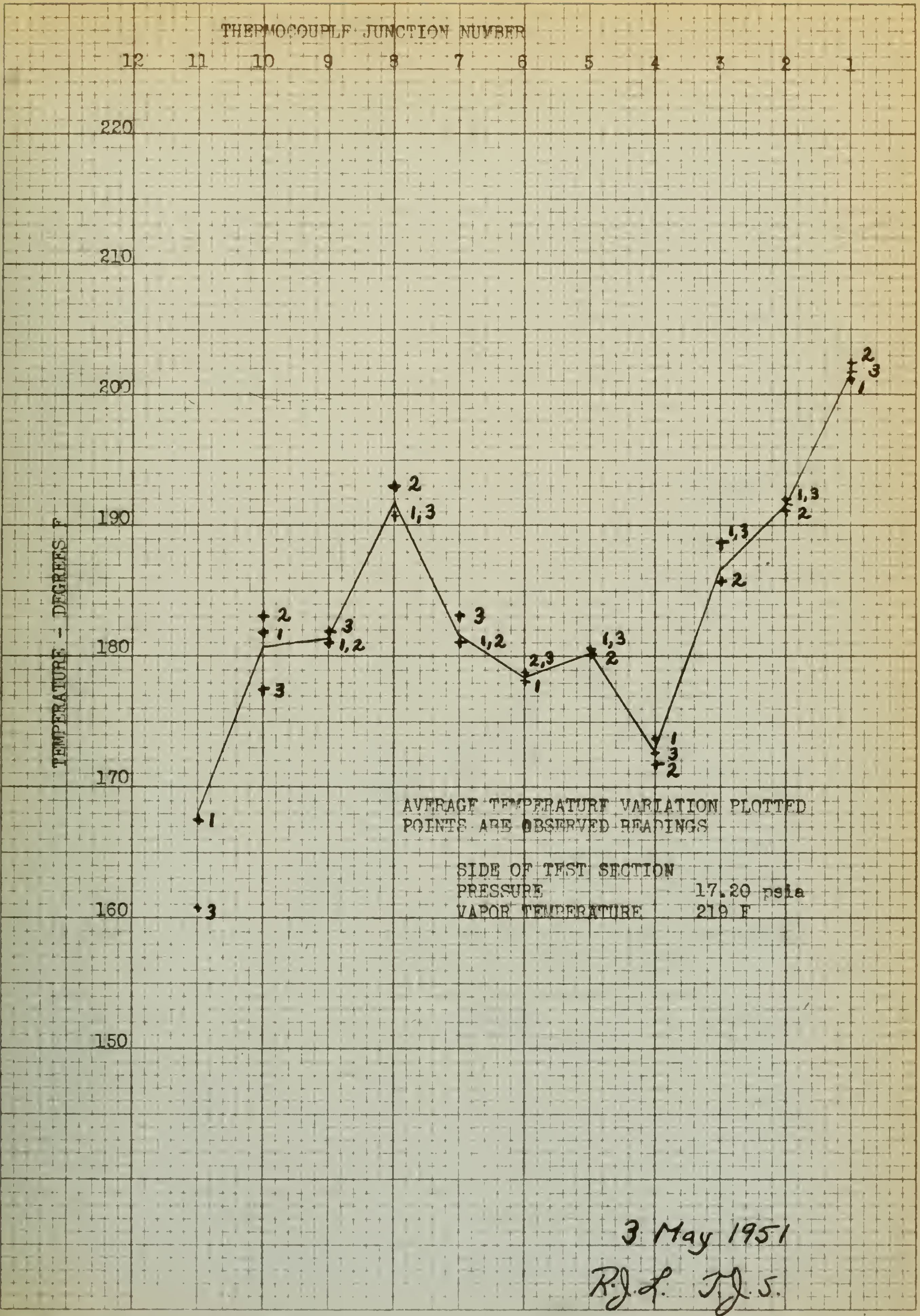
220
210
200
190
180
170
160
150
AVERAGE TEMPERATURE VARIATION PLOTTED
POINTS ARE OBSERVED READINGS
TOP OF TEST SECTION
PRESSURE 17.15 psia
VAPOR TEMPERATURE 219 F
3 May 1951
R.J.L. T.J.S.

FIGURE V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6
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4
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2
1
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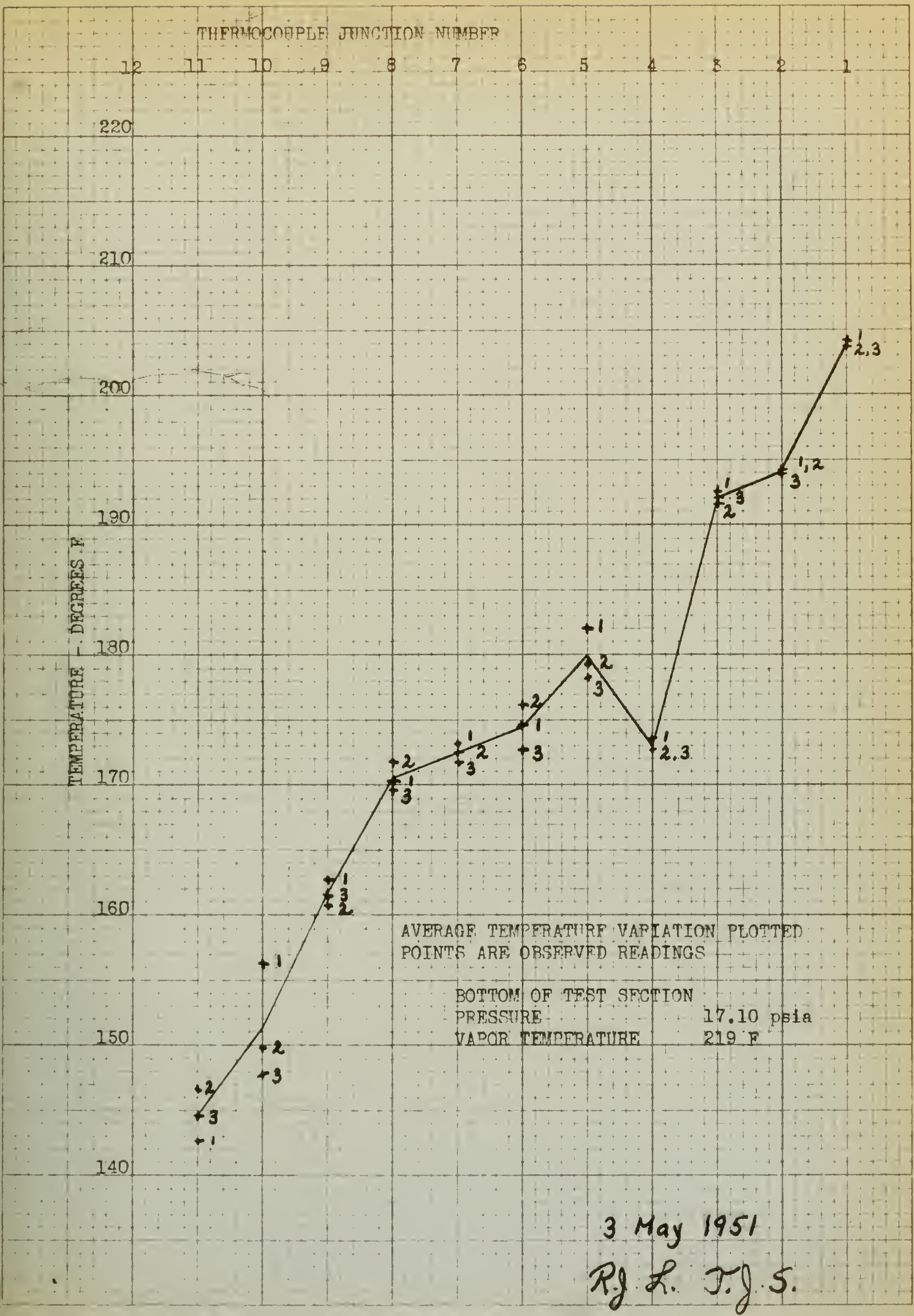
210
200
190
180
170
160
150
AVERAGE TEMPERATURE VARIATION PLOTTED
POINTS ARE OBSERVED READINGS
SIDE OF TEST SECTION
PRESSURE 17.20 psia
VAPOR TEMPERATURE 219 F
3 May 1951
R.J.L. T.J.S.

FIGURE VI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140
AVERAGE TEMPERATURE VARIATION PLOTTED
POINTS ARE OBSERVED READINGS
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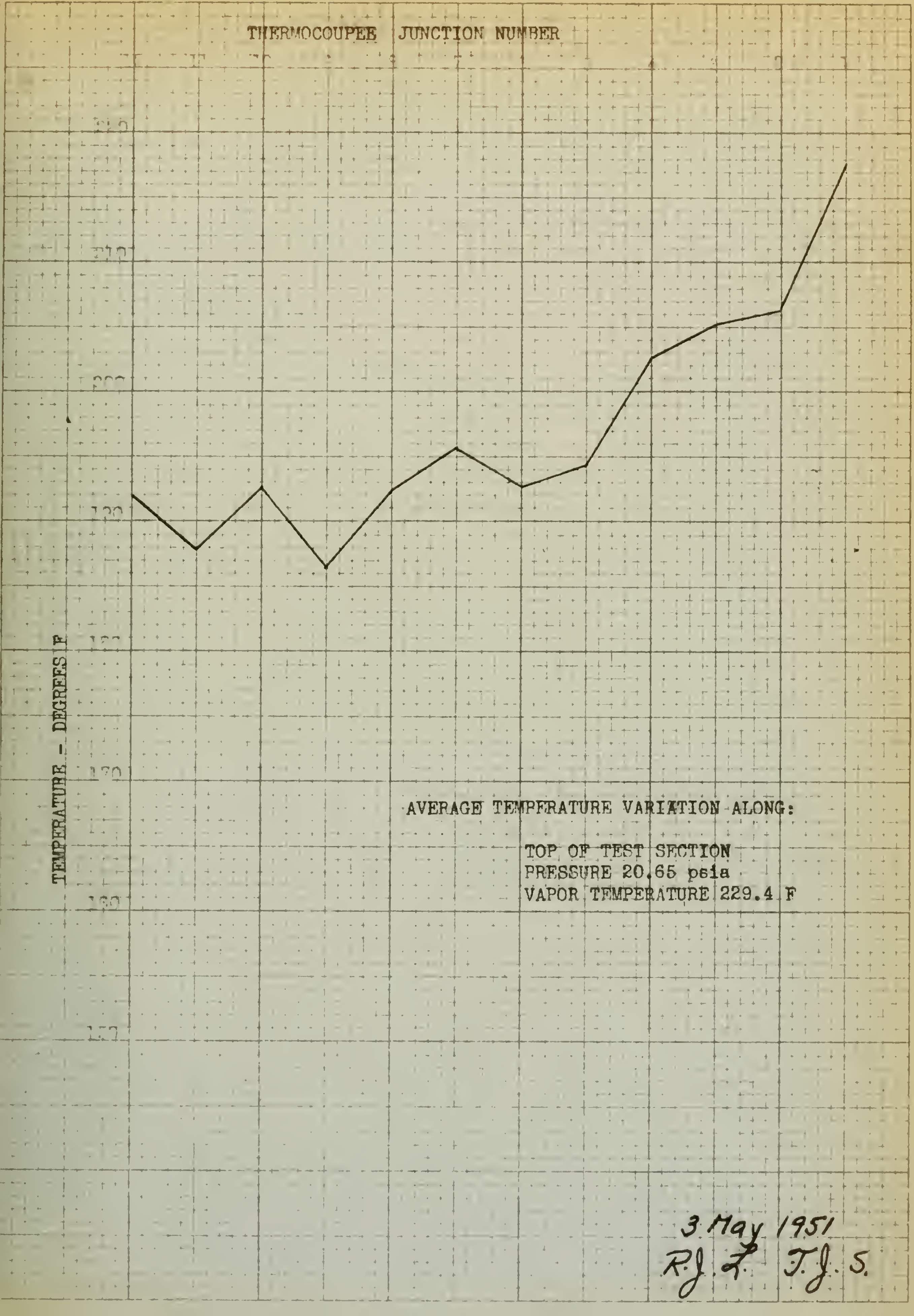
VAPOR TEMPERATURE 219 F
3 May 1951
R.J. L. T.J. S.

FIGURE VII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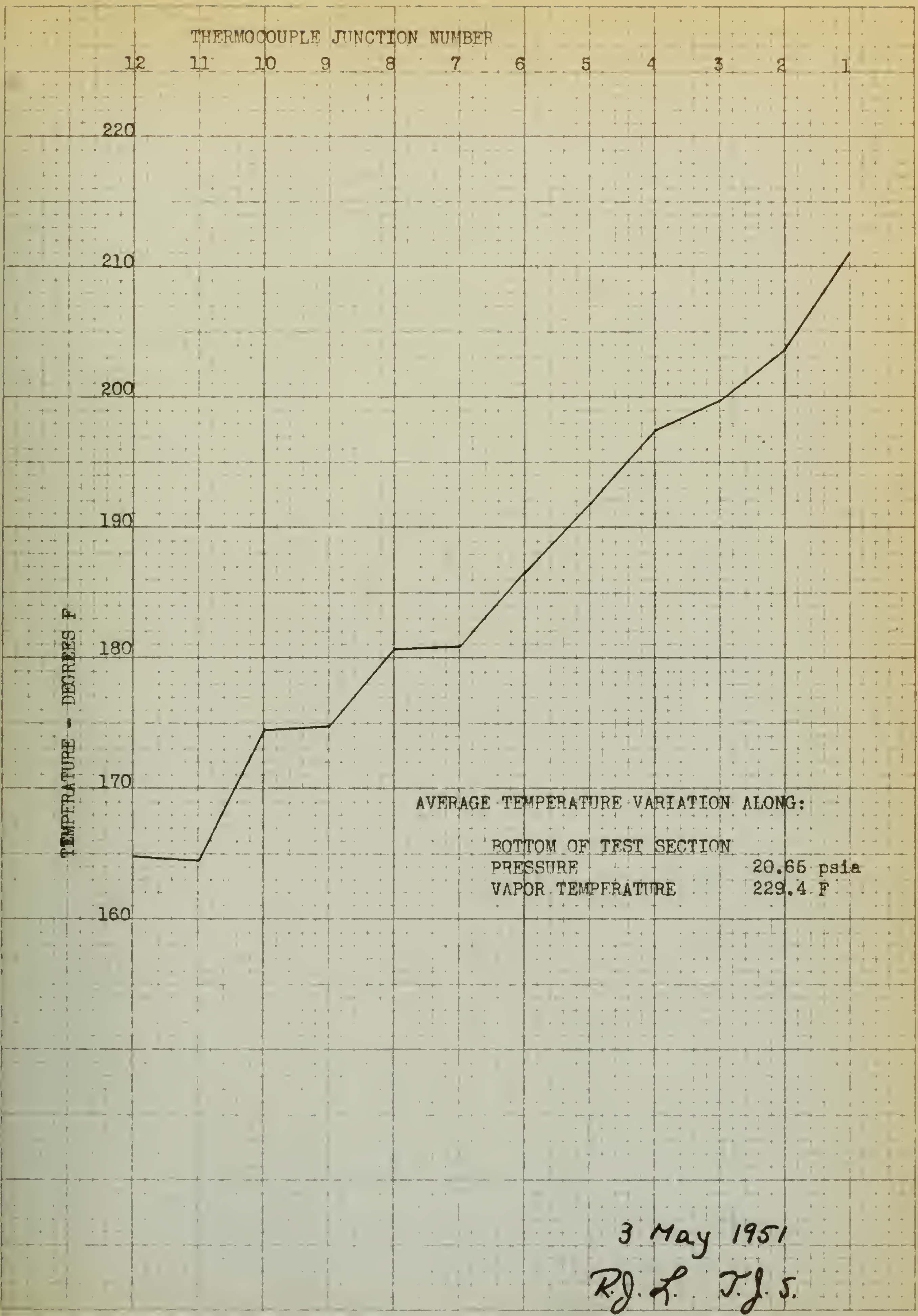
AVERAGE TEMPFRATURE VARIATION ALONG:
TOP OF TEST SECTION
PRESSURE 20.65 psia
VAPOR TEMPERATURE 229.4 F
3 May 1951
R.J.L. T.J.S.

FIGURE VIII

3 May 1951

R.J.L. T.J.S.

FIGURF IX



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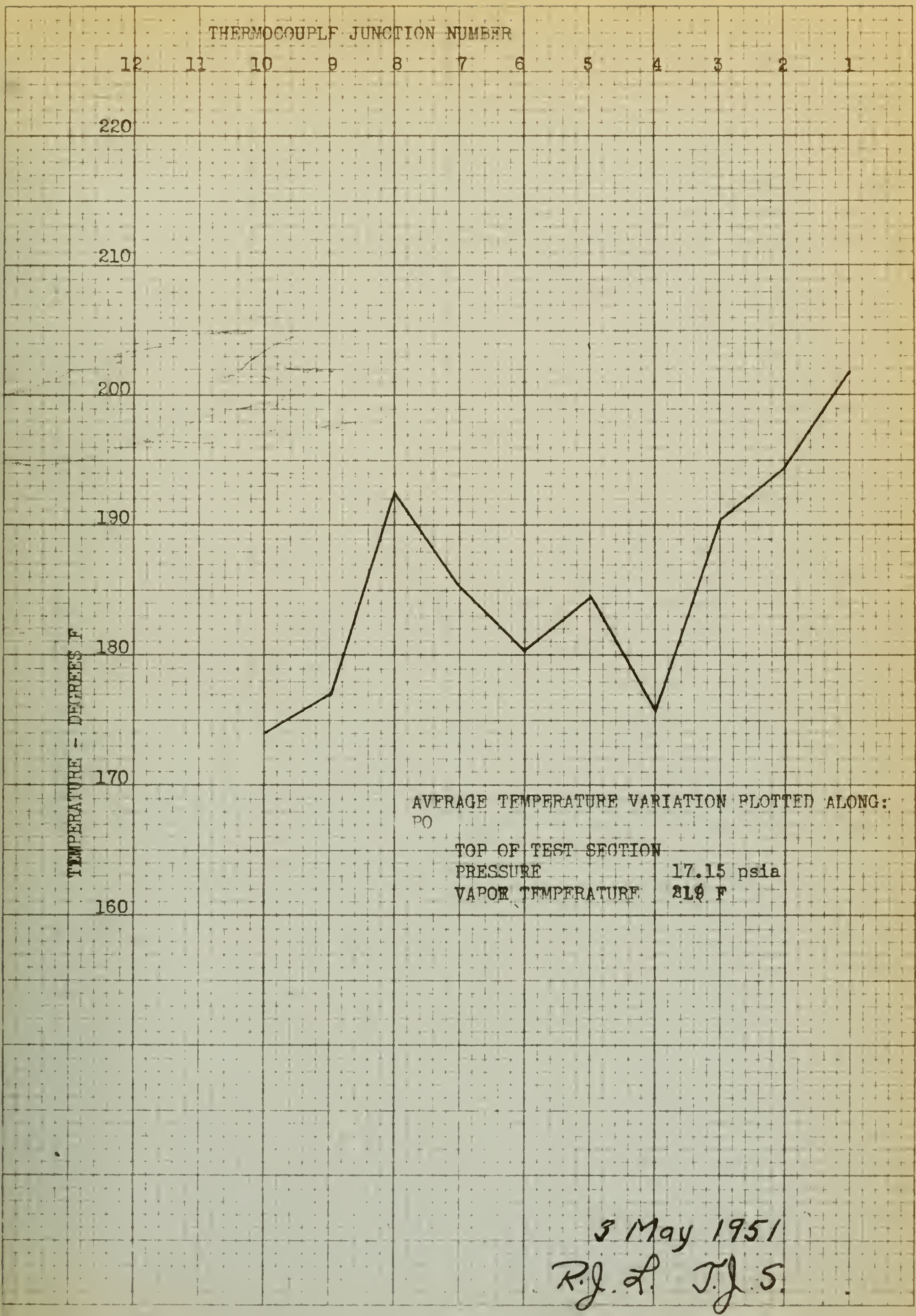
160
TEMPERATURE - DEGREES F
AVERAGE TEMPERATURE VARIATION ALONG:
BOTTOM OF TEST SECTION
PRESSURE 20.65 psia
VAPOR TEMPERATURE 229.4 F
3 May 1951
R.J.L. T.J.S.

FIGURE X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170
160
TEMPERATURE - DEGREES F
AVERAGE TEMPERATURE VARIATION PLOTTED ALONG:
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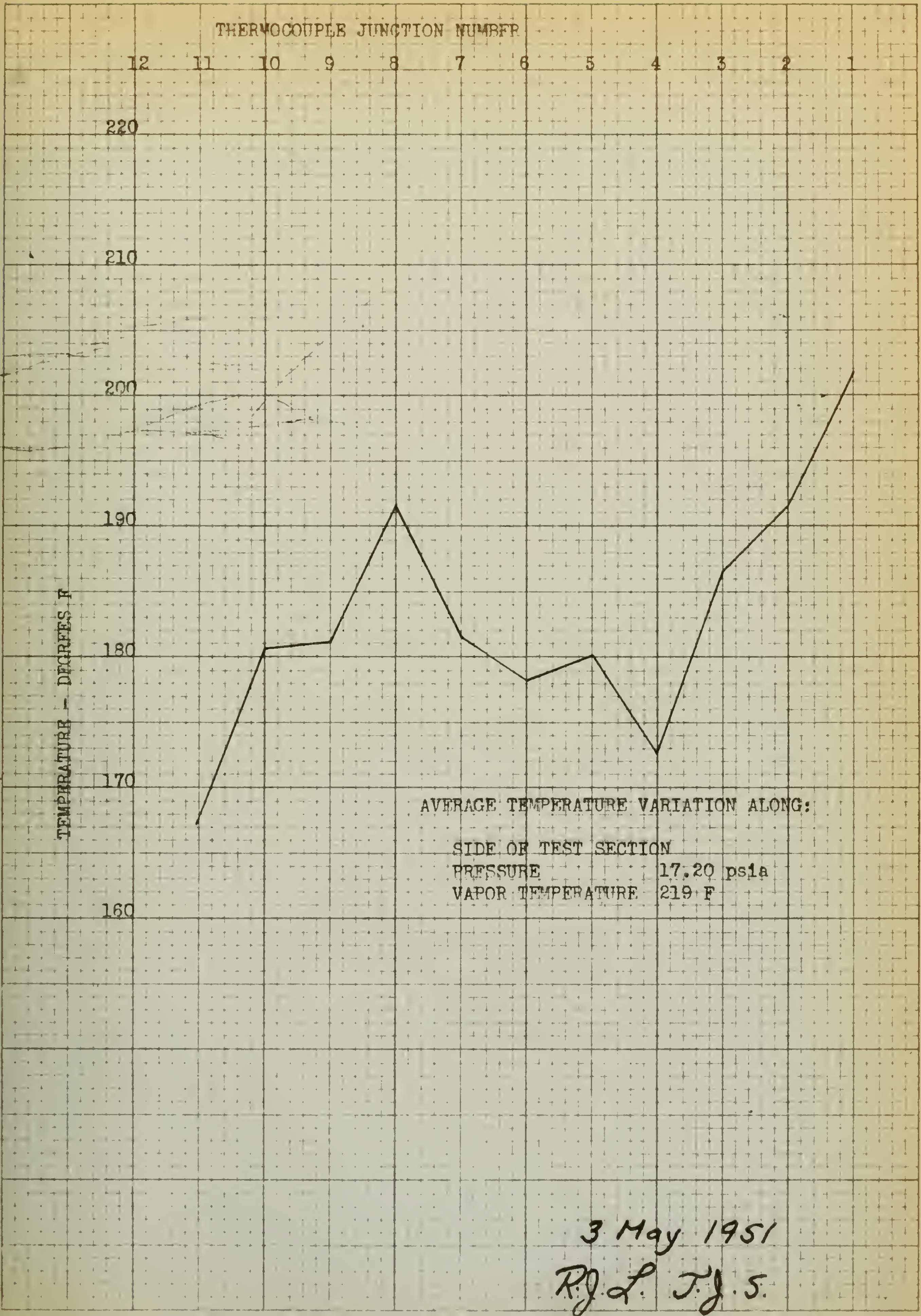
TOP OF TEST SECTION
PRESSURE 17.15 psia
VAPOR TEMPERATURE 219 F
3 May 1951
R.J.L. T.J.S.

FIGURE XI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TEMPERATURE - DEGREES F
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PRESSURE 17.20 psia
VAPOR TEMPERATURE 219 F
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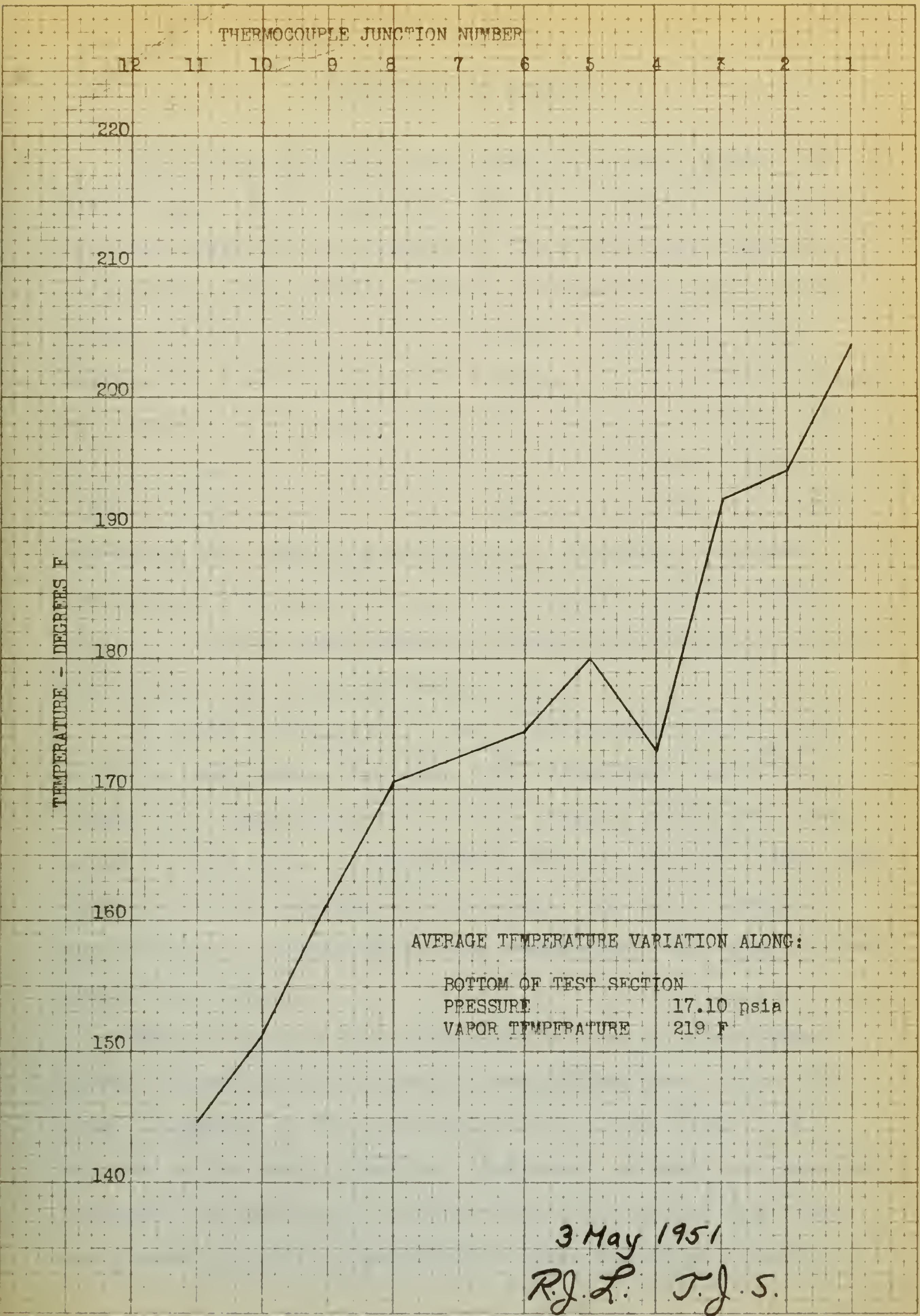
R.J.L. T.J.S.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AVERAGE TEMPERATURE VARIATION ALONG:
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PRESSURE 17.10 psia
VAPOR TEMPERATURE 219 F
3 May 1951
R.J.L. T.J.S.

DISCUSSION OF RESULTS

The work of previous investigators on this problem has been meager. It is considered that the principal contribution of this thesis to the solution of the problem has been one of definition of the problem itself, and demonstration of techniques which will aid in the solution. Definition of the problem has been stated in the Summary. It is to find a type of experimental apparatus which will provide in nature, conditions within the tube approaching the conditions imposed by a Nusselt type analysis of the condensing process. One must provide a heat absorbing medium which approaches constant temperature even though it is receiving the heat of condensation. One also must provide a horizontal tube in which the velocity of the vapor approaches zero.

By rapid circulation of the cooling water and by using a single pass across the tube, well distributed along the length, the temperature rise of the cooling medium was minimized in this work. A temperature rise of only 15° F has been achieved and is considered to be good. By use of greater quantities of cooling water, its temperature rise could be reduced even further.

These observers have discussed the use of a different scheme for effecting a constant temperature heat sink. The idea, adaptable to the equipment used in this thesis, is to use ice as the heat absorbing substance. If the heat transfer surface were immersed in an ice water bath and if the entire bath were agitated mechanically, a heat absorbing medium of

[illegible]

[illegible]

[illegible]

[illegible]

constant temperature would be obtained. This scheme has the disadvantage that the cold wall temperature would be very low indeed. So that there would not be an excessive temperature differential between the condensing vapor and the cold surface (leading to high rates of heat transfer and excessive velocity of the vapor), one would have to reduce the temperature of the condensing vapor. For steam vapor this would lead to extremely low pressures, not handily achieved. But for other vapors, having lower saturation temperatures at atmospheric pressure, the use of ice for cooling should be borne in mind.

As regards vapor velocity conditions, the equipment used in this thesis failed to meet the requirements of the Nusselt assumptions. The velocity of the vapor should approach zero. The velocity of the vapor in the equipment used in this work approached 180 feet per second, a very excessive velocity. What was wrong?

In the first place, the test section was too long. It had been made so deliberately to minimize heat loss at the ends of the section. However, end effects were discovered to be of very minor consequence. All of the vapor being condensed along the length of the tube has, of course, to pass through the entrance cross section. For a cross section of any given area, the longer the test section the greater the velocity of the vapor at the entrance.

Secondly, two constants, arbitrarily imposed upon the equipment, increased the velocity. They were: (a) the temperature of city water in the mains, and (b) the temperature of

condensing steam at atmospheric pressure. The large temperature differential existing between these two gave rise to a large condensation rate, large vapor volume entering the section, and high velocity of vapor.

Certainly then, future modifications to this equipment ought to incorporate one or both of these features; a smaller temperature differential between the cooling medium and the condensing steam and/or a shorter length of tube as compared to diameter. A smaller temperature differential might be achieved either by heating the cooling water, by reducing the pressure inside the tube below atmospheric, or both. A test section having a length on the order of 2 feet and a diameter on the order of 1 inch is recommended.

Most of the available time for the thesis was spent in construction of the apparatus and in modifications to it as found necessary. Experimental tests were conducted at only two vapor pressures. It would be unwise to extrapolate from such little data to any major quantitative conclusions. The results of the tests do, however, deserve comment.

FIGURES I, II, and III

Figures I, II, and III indicate the variation in temperature along the length of the test section for the high pressure run. The position of the thermocouple junctions along the length of the tube is illustrated in Figure XVI. Thermocouple junction number 1 is closest to the entrance of the tube, and thermocouple number 12 is closest to the end through which the condensate was removed. There were three observations made for each point. The straight line segments

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[illegible]

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[illegible]

connect the average values of these three observations. For the most part there is little spread between the three, so that the average is a representative value for temperature at the point.

In Figures I, II, and III, which are for temperature distribution along the top, side, and bottom of the tube respectively, it should be noted that there is a steady decline in temperature from the entrance end of the tube to the exit end. A high temperature of the outer tube wall means that there is small thermal resistance between that point and the condensing steam. The Figures show that the film is very thin near the entrance to the tube, and that the liquid film progressively increases in thickness toward the far end of the tube. The effect of the vapor velocity is illustrated. Thermocouple junction 1 indicated the highest temperature of all because the entering steam vapor wiped off any condensate which formed. For the top and sides of the test section, as shown in Figures I and II, the velocity of the vapor seems to have had little effect beyond junction number 6. For junctions 6 through 12 the temperature remained rather constant, indicating that the film thickness was not seriously distorted by the vapor velocity beyond the mid-length of the test section. On Figure III, however, the temperature is seen to decrease steadily from junction number 1 all the way through to the exit end of the tube at junction number 12. Evidently, the thickness of the water film undergoes a continual build up along the length of the test section, being deepest at the point where it flows out. If the velocity of the vapor had

no effect on the liquid the opposite would occur, the deepest film would be found at the entrance cross section, providing the hydrostatic head to cause flow of the condensate to the outlet end.

The relative thicknesses of the films on the top, bottom and sides of the test section are illustrated. If the Figures I, II, and III are held together and interposed between a strong light source, one can observe that the temperatures on the top of the tube are higher in every case than the temperatures on the sides of the tube. The temperatures on the sides of the tube, likewise, are higher than those for corresponding cross sections on the bottom of the tube. Such results indicate the build up of the film from the top around to the bottom of the tube.

How much of this temperature difference was due to change in film thickness? How much of it was due to the fact that the cooling water was brought in at the bottom of the test section, flowed around it and was withdrawn from the top of the condenser shell 15° F hotter than when it came in? These questions remain unanswered. They point up the necessity of minimizing the temperature rise of the cooling medium. Say, for example, that the expected temperature difference between the side and the top of any cross section, due to a difference in film thickness at the two points, is on the order of 5° F. If the cooling water undergoes a temperature rise in excess of that amount, say 7°, by reason of heat absorbed from the tube, then the temperature distribution around the circumference is distorted. Such a distortion would affect observed

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temperatures whether they were taken on the outside of the tube surface, as was done in this thesis, or the inside of the tube, as was done by Jakob.

The composition of the tube wall is an important variable. In this thesis, copper tubing was chosen in order to minimize the thermal resistance of the tube wall. Since the total resistance to heat flow in the test section consists of two thermal resistances in series, the film of condensate liquid and the tube wall, it becomes desirable to make the resistance of the wall as small as possible. The resistance of the liquid film will be a greater proportion of the total and it may be measured more accurately. This is the thinking which led to the selection of copper for the tube wall material.

However, if the temperature gradient around the circumference of the tube wall were high, as a natural condition of the film thickness, the choice of high conductivity copper is poor. The copper tube wall would tend to even out temperature differentials from point to point in the circumference.

FIGURES IV, V and VI

Figures IV, V, and VI show the temperature gradient along the length of the test section for the lower pressure run. The general comments which were made above concerning increase in temperature of the cooling medium and the composition of the tube wall apply to these results as well. Some additional comments are in order.

Prior to the experimental test runs at the lower pressure the second manometer was installed on the apparatus, on the

exit end of the test section. For an unknown reason this modification to the equipment caused trouble at the exit end. The small cock, installed to bleed non-condensable gases from the test section, began spouting water. When it was shut off, an air pocket was formed at the exit end and very low temperatures, on the order of 90° F, were indicated by the thermocouple junctions at that end. These temperatures, known to be erroneous, were not plotted on the graphs.

Subsequent to the experimental runs an inspection was made of the apparatus to find out why the indicated temperatures on junction number 4 should be so much lower than the temperatures at adjacent junctions. It was noticed that water had entered the glass tube at the reference junction for thermocouple number 4 due to the fact that the glass tube was broken. It is considered that this water resulted in an erroneous reading for thermocouple number 4. The average temperatures plotted at the other points exhibit much the same characteristics as those which are shown on Figures I, II, and III for high pressure run.

It is interesting to note that in both of the runs, using average pressures and temperatures, that the vapor velocity was essentially a constant. This undoubtedly is a rare coincidence, but it is believed that the vapor velocity for the apparatus used should remain in about the same neighborhood. In the formula $V = \frac{v\ w}{A}$, A is constant, v is a function of the steam conditions and increases as the pressure decreases (this is assuming a saturated vapor), and w, the weight of condensate, depends on the Δt drop and v. Hence, as the

pressure drops the temperature of the steam decreases and v increases. Since the cooling water is at a constant temperature essentially, the Δt drop is lower and this coupled with the higher v gives a smaller amount of condensate. This results in two variables changing in opposite directions which tend to give a somewhat constant numerator divided by a constant denominator. Thus the vapor velocity remains essentially constant.

As seen by the computations, the overall U values differ, being lower in the lower pressure case. This is brought about because the total Q is smaller in the low pressure case and the Δt drop is the same. Since the vapor velocity was found to be almost constant for both cases the difference which caused the different U values is the turbulent layer of the film. If it can be assumed that the laminar layer is essentially constant in thickness, the only remaining resistance is the turbulent layer. The turbulent layer increases in the high pressure case because v is smaller and as a consequence the weight of condensate increases, which thickens the turbulent layer. The turbulent layer increases heat transfer, and thus a higher value of U should be obtained for the high pressure case than for the low pressure case, which was found to be true. The foregoing is based on the premise that the film does not get so large as to materially clog the pipe.

The pressure drop was measured in the last run but was not in any of the calculations since it did not prove applicable to the problem.

The high vapor velocity encountered in this problem and the consequent reversal of the slope of the film in the tube obviously makes the theoretical Nusselt approach to the problem relatively inapplicable, since the Nusselt assumptions is that the vapor velocity be negligible. The natural build up of a water head to create flow in the horizontal tube is not obtained since the vapor velocity is not negligible. The film inside the tube is largely turbulent instead of laminar as would be predicted by the Nusselt theory. With this setup and apparatus it is possible to get negligible vapor velocity by increasing the temperature of the cooling water up to near the 200° F point. However, this poses a difficult problem, since facilities for heating such a great quantity of "cooling water" are unavailable. If it were possible to heat the cooling water this high, the result would be a small amount of condensate and negligible vapor velocity. To draw a simile with actual practice to which this thesis is pointed, in modern refrigeration plants of 5 tons capacity and above, the flow of refrigerant is considerable. Realizing that at the high pressures at which the refrigerant leaves the compressor the v of the refrigerant is small and in passing through the condenser, it is felt that the velocity of the vapor will not be negligible. Hence, the vapor velocity will have to be considered in order to have a realistic approach to this complex problem. This addition will materially disturb the reasoning and call for a complete revision of the formulae propounded to date.

CONCLUSIONS AND RECOMMENDATIONS

The apparatus as presently installed has several points which are of dubious value and the following corrective measures should be taken prior to operating this apparatus again.

1. The entrance thermocouple should be removed from its present position and placed in the downcoming section of the steam line just before the test section. This should be done due to the possibility of accumulating water in the $1\frac{1}{4}$" line and also the possibility of getting an air pocket at this junction.

2. Considerable difficulty was experienced during our second run with the pet cock installed to act as an air vent. Either the manometer should be installed in the test section line or the pet cock placed on a higher riser. This should be done because the installation of the manometer in the bronze elbow created great difficulty. The difficulty being that whenever the pet cock was cracked to let a whisper of steam out, a considerable quantity of condensate also came out. It is felt that this situation could be alleviated by installing the manometer in the line and putting the pet cock back in its original place in the bronze elbow.

3. The discharge end of the test-section-water plenum should be rethreaded. The present threads are cocked and when the test section is installed it experiences a bending, which disturbs the horizontal requirements. This bending of the test section is aggravated when the test section has steam in it.

4. The six-foot section is believed to be too long for sufficient stiffness to support its own weight. It is believed that a two or three foot section would be of sufficient stiffness and also be more compatible with condenser tube lengths used with refrigerants today.

5. An electrical superheater must be installed at the steam inlet in order to insure a sufficient degree of superheat.

6. In the course of taking reading, during the second run station 4 was obviously incorrect. It was found that the reference junction for four had a cracked glass tube. All the tubes used should be rechecked to insure no more of this occurs.

7. With the present setup, an average cooling water temperature rise of 18-20 degrees is experienced. If possible the inlet water line should be increased to 1" pipe in lieu of the presently installed ½" pipe. The volume of cooling water will then be increased approximately fourfold, and the rise of the cooling water should be in the neighborhood of 5 degrees.

8. The entrance manometer should be placed in the test section instead of its present position in the 1¼" line. The venturii effect of the reduction in pipe size from 1¼" to 3/4" is not known, and in order to get the pressure drop in the test section the pressure at both ends of the test section must be accurately known.

9. The installation of a pitot tube in the 1¼" line would permit determination of the velocity of vapor at the

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[illegible]

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entrance to the test section.

10. If this apparatus is to be discarded and a new shorter test section employed, it is recommended that the thermocouples be put on the test section in a different manner. The manner is one used in the chemical engineering department. Holes are drilled into the pipe tangentially and the thermocouples installed in the holes. This would be a great deal more practical and also a great deal more rugged than the present system employed of soldering the thermocouples in grooves.

APPENDIX

DETAILS OF PROCEDURE

As has been stated in the Introduction, the investigation of film type condensation seems to consist of two separate parts. One is an attempt to provide conditions within the experimental equipment which will approach the limiting assumptions of the Nusselt theory. The other is to approximate conditions found in commercial condensers. The difference between these two parts has been previously discussed. At the outset of this investigation it was decided that the former problem would occupy the attentions of these observers.

One assumption of the Nusselt theory is that the temperature difference between the condensing vapor and the colder condensing surface is constant at all points. The assumption requires that the cooling medium remain at the same temperature after it has absorbed heat given up by the condensing vapor. Such a condition is not found in nature. One would need an infinite heat sink at the wall to effect it, but it can be approached in practice. Heat transfer in a counterflow condenser would not approach the above stated condition. It was decided that cross-flow of the cooling medium, with one pass, might approximate it. Accordingly, the heat transfer surface was immersed in the cooling water which was fed into the cooling shell through twelve inlet nipples on the bottom and out through the same number on the top of the shell. To provide for equal flow of the cooling water through each of the twelve nipples it was necessary to

install two plenums, one feeding the inlet nipples and the other receiving the cooling water emerging from the outlet nipples. Then in order to further insure equal pressure drop across any pair of nipples, the cooling water was introduced on one side of the inlet plenum and withdrawn from the opposite side of the outlet plenum.

The condensing shell and the two plenums were constructed of 5 inch steel pipe, blank flanged on the ends. The nipples were of 3/4 inch pipe. It was planned to silver solder the nipples in place at both ends, but because of the high temperature of the soldering process and differential expansion of the shell and plenums the nipples would not hold in place. Rubber hose was finally used to couple the nipples together. So much for the cooling water circuit.

It was felt that the second limiting assumption of the Nusselt theory could be met in practice by throttling the inlet steam, restricting the flow so that the velocity of the vapor would not be excessive. The equipment which was constructed failed to meet this qualification. Comments on the vapor circuit follow.

Considerable discussion was given to the choice of steam supply. At first it was thought that steam might be taken from steam mains of the Institute, but no adequate outlet existed in the Heat Measurements Laboratory where the test equipment was to be located. Also, it was felt that the steam mains were probably contaminated with oil which would have to be filtered out. An ideal source of steam would have been a clean boiler fed with distilled water. This idea was rejected

because of the size of the constructional work which would have been required in building a boiler to give evaporation rates sufficient to feed this test section. A compromise was made in the choice of a small heating boiler already existing in the Heat Measurements Laboratory. This boiler which was finally used provided steam at pressures up to about 5 p.s.i. gage and, since it was automatically gas fired, a means of adjusting the outlet pressure.

For the low rates of steam supply originally anticipated a 1/2 inch supply line, suitably trapped and vented, was run from the boiler to the test section. The boiler provided steam which was nearly saturated at these low rates of flow, but to insure saturation an electrical resistance superheater was installed on the steam supply line. The energy input of the superheater was controlled by means of a Variac transformer.

The test section must pierce the blank flanges on the ends of the cooling water shell. To permit this two stuffing boxes were machined and installed in the flanges. The face of the stuffing boxes was scribed with marks at 30° angles. The test section of copper tubing was scribed with a single line along which the thermocouple junctions were mounted. By this means it was considered that the copper test section could be rotated to any angle at known intervals.

Before passing on to a discussion of the condensate piping, it must be noted that the 1/2 inch steam supply line was found to be too small. When initial tests were run on the equipment just described, it was found that the restriction to steam flow offered by the 1/2 inch supply main disturbed operation

of the whole apparatus. The steam supply line subsequently had to be increased to 1 1/4 inch diameter. The reasons for this are described below.

With the 1/2 inch steam supply in place, the rate of cooling water had to be made very low indeed. If the rate of cooling water were increased, steam would condense inside the test section so rapidly that a vacuum would be formed there and the condensate would not flow out to atmosphere. When the cooling water rate was reduced to the point where a positive pressure above atmosphere could be maintained on the steam side of the test section, the increase in temperature of the cooling water was approximately 100° F. These unfortunate conditions were obtained even though the boiler pressure was held at its maximum, about 5 p.s.i. gage. The flow rate in the 1/2 inch steam line was such that a 5 p.s.i. pressure drop was occurring between the boiler and the entrance to the test section.

It is necessary to digress here and to point out that there are no independent variables in this work. What is done to one part of the heat transfer equipment affects all of it. These observers had assumed that they were to hold the entering velocity of the steam vapor low and that a small steam supply line would be sufficient to carry it. But the cooling water temperature, taken from city water supply mains, is very cold compared to steam vapor at atmospheric pressure. With such a large temperature differential high rates of heat transfer are indicated, and in a long tube as was installed here, it resulted in either excessive rise in the temperature

of the cooling water or excessive velocity of the entering steam vapor.

The diameter of the steam supply main was increased to 1 1/4 inch pipe. This modification to the equipment resulted in a sufficient steam supply to the test section so that a positive pressure was maintained there. The rate of cooling water could be increased to the maximum possible which the water supply main could carry. However, then the steam vapor rate became so high that the effect of vapor velocity became all important.

The condensate piping system will now be discussed. From the exit end of the copper test section, the condensate was led to a small cylindrical, welded tank named the saturation chamber. A needle valve was inserted in the condensate line to control the flow of condensate. The condensate piping was next led through a cooling coil immersed in a large bucket of cold water. This coil effected cooling of the condensate so that it would not flash to the vapor state upon being brought out to atmospheric pressure.

Placing the thermocouples in place was a delicate task. The copper test section was scribed with a line along the outer surface. Circumferential grooves with a depth just great enough to accommodate the #28 thermocouple wire were cut by machine at 6 inch intervals along the length of the tube. The copper and constantan leads were soldered in place so that the thermocouple junction would lie just on the scribed line. Because of the high thermal conductivity of

the copper tubing care had to be taken in the use of the soldering torch so that when one junction was being soldered, the junction adjacent to it would not melt out. It is considered that on such copper tubing junctions could be placed no closer than the 6 inch interval used with this equipment, unless some special clamps were used to hold several junctions in place at one time.

To insure the electrical insulation of the thermocouple leads, which must necessarily be run immersed in the cooling water, three coats of polystyrene were applied to each wire over the entire wetted length. The temperature of the cooling water at even a short distance from the tube surface is very cold compared to the temperature existing at the junction. To avoid the disturbing effect of conduction of heat along the length of the leads, especially the copper one, both leads from each junction were wrapped tightly around the copper tubing in the groove cut for that purpose.

The leads tagged with numbers, brought away from the tube surface and led to the outlet end of the condenser shell. At each junction, of course, two leads were added to the group of strands, so that at the final end of the tube 24 individual leads had to be brought out through the blank flange. The leads were withdrawn from the shell in between two rubber gaskets fitted at that end of the shell.

Mercury manometers were installed at each end of the test section. A small cock was installed at the outlet end of the test section so that non-condensable gases could be bled from it. The cock was cracked open minutely so that a

whisper of steam emerged from it.

Two new spools of thermocouple wire were purchased for the equipment. It was realized that, to minimize the effect of thermal conduction in the leads, the smallest practical diameter wire was desirable. A wire size #30 was sought, but this being unavailable, #28 wire was used. The wire was calibrated at the steam point only. A linear deviation between the steam point and the ice point was assumed. Three separate calibration tests, giving consistent results, were run. From data published by the U. S. Bureau of Standards, an expanded graph was drawn showing the relation between Observed Millivolts and Temperature in Fahrenheit degrees. The indicated temperature of the thermocouples could thus be readily obtained.

To support the condenser shell and the two plenums, an A-frame structure was built of 1 1/4 inch angle iron. Suitable cross pieces were fitted so that each plenum and the shell would be supported separately. Four bolts were installed in the base of the A-frame, one at each corner. By means of these bolts, raising or lowering them, the copper test section was placed in a horizontal line. A length of rubber hose was fitted with two pieces of glass tubing, one inserted in each end. The hose was filled with water, providing a long water level. The ends of the test section which projected outside of the condenser shell were adjusted to the same vertical height by means of the water level and the adjusting bolts.

Visual observation of the test section after this had been done sufficed to show that the whole of the copper tubing

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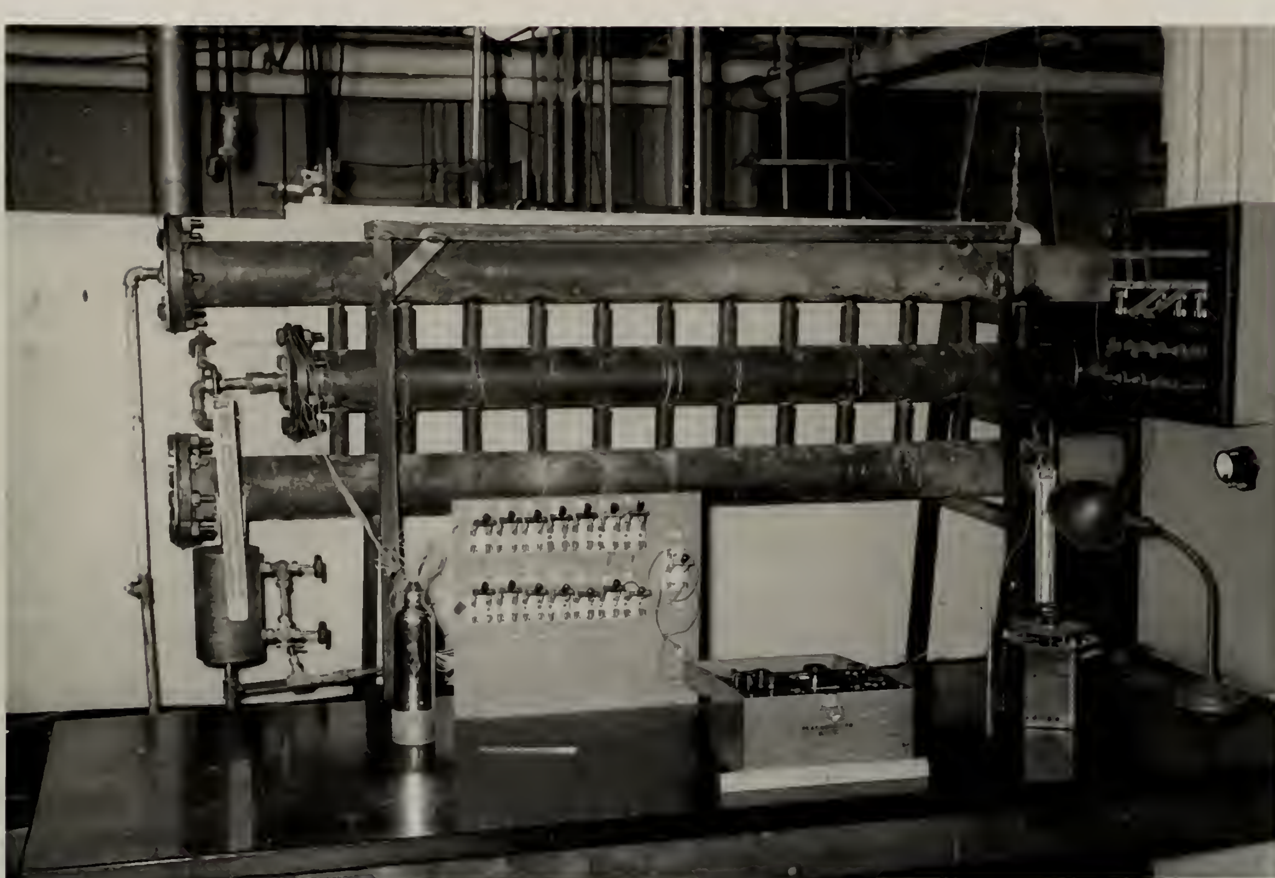
[illegible]

was not in perfect horizontal alignment. This was attributed to two causes. First, the section modulus of copper tubing is not great. The six foot length of tubing simply supported at the ends, bends like a beam under its own weight, creating a hollow in the center where condensate may build up to some depth. Secondly, the thread which had been cut on the 5 inch pipe was cut at a skew angle. Therefore, the flange was slightly skewed, and the stuffing box at the outlet end of the test section was cocked. The test section fit into the stuffing box was a tight one. Play was taken up in the packing, but the net effect was a deviation from the horizontal along the length of the test section.

FIGURE XIII



Front View of Test Equipment

The condenser shell is in the middle, inlet plenum below it and outlet plenum above it. The test section may be seen projecting through the blank flange of the condenser shell. Note the manner in which thermocouple leads are brought through the flange at left.

FIGURE XIV

Rear View of Test Equipment

The boiler is shown at left. The condensate system is shown coming out of the condenser shell, down through the saturation chamber, through the needle valve, and into the barrel in the middle where the condensate is subcooled.

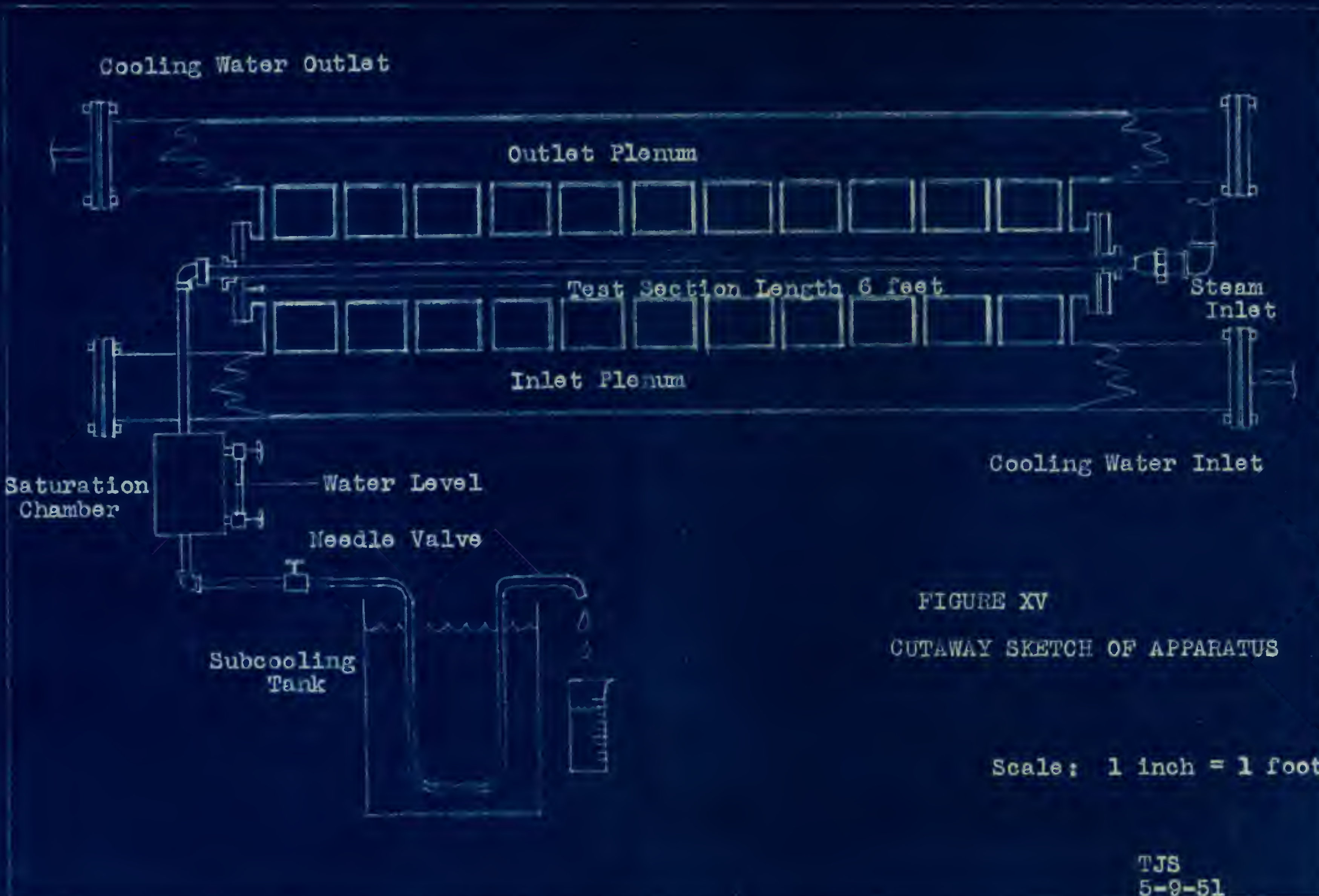


FIGURE XV

CUTAWAY SKETCH OF APPARATUS

Scale: 1 inch = 1 foot

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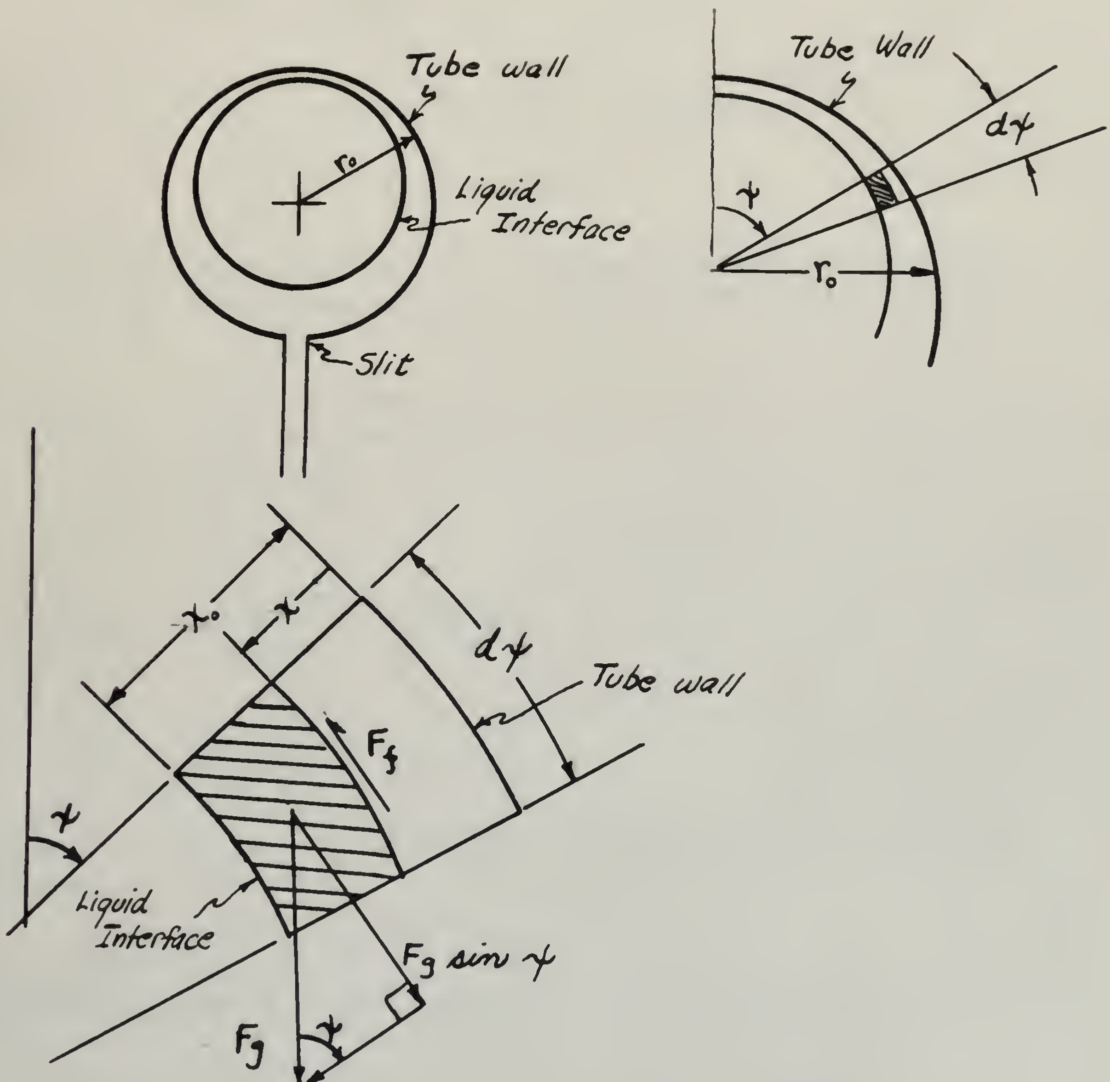
NUSSELT TYPE DERIVATION FOR PREDICTING HEAT TRANSFER COEFFICIENTS OF VAPOR CONDENSING INSIDE HORIZONTAL TUBES

Assumptions:

(a) the vapor is pure and saturated
(b) the tube wall temperature remains constant
(c) the vapor has no velocity relative to the interface
(d) heat flows through the condensate only by conduction
(e) the condensate flows viscously down the wall surface under the action of gravity alone
(f) two dimensional representation of the condensation including the presence of a hole at the bottom of the tube to carry off the condensate which is formed.



It is first desired to find the relation between the thickness of the liquid film, χ_o, and the variable angle ψ. Consider the shaded element shown in the sketches. As it flows down the tube wall it is acted upon by two forces, the force of viscous friction and gravity. Since there must be equilibrium:

$$F_f = F_g \sin\psi \tag{1}$$

$$\mu (r_o - \chi)\, d\psi \, \frac{\partial v_t}{\partial \chi} = \gamma\, (\chi_o - \chi)(r_o - \chi)\, d\psi \sin\psi \tag{2}$$

$$\int_0^{v} dv_t = \frac{\gamma}{\mu} \sin\psi \int_0^{x} (\chi_o - \chi)\, d\chi \tag{3}$$

The limits of integration may be so set because it is assumed that the velocity of the liquid is zero at the wall. Integrating,

$$v_t = \frac{\gamma \sin\psi}{\mu} \left(\chi_o \chi - \tfrac{1}{2}\chi^2 \right) \tag{4}$$

one finds the tangential velocity of the liquid, v_t. The mean velocity of the liquid, v_m, across any section at the variable angle ψ is given by:

$$v_m = \frac{\int_0^{\chi_o} v_t \, d\chi}{\int_0^{\chi_o} d\chi} \tag{5}$$

$$v_m = \frac{\gamma \sin\psi}{\mu\, \chi_o} \left[\tfrac{1}{2}\chi_o^3 - \tfrac{1}{6}\chi_o^3 \right] \tag{6}$$

$$v_m = \frac{\gamma \sin\psi}{3\mu} \cdot \chi_o^2 \tag{7}$$

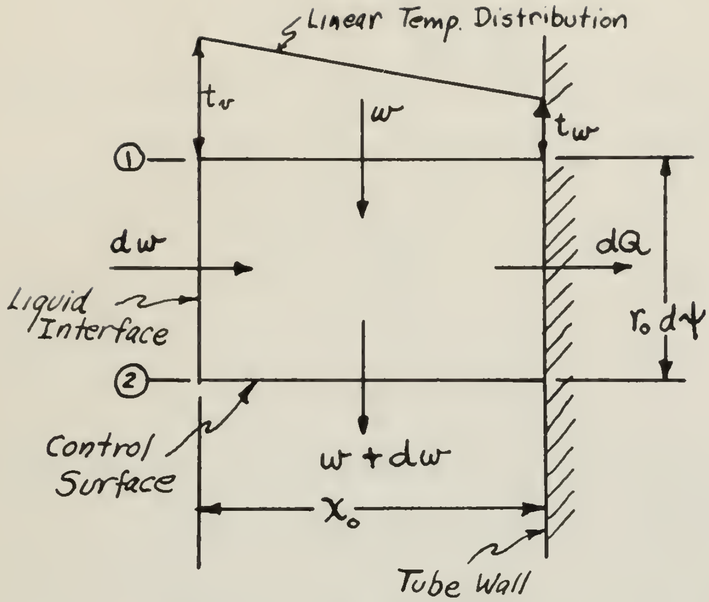
The mass rate of flow of condensate, w, across a section of width χ_o must then be:

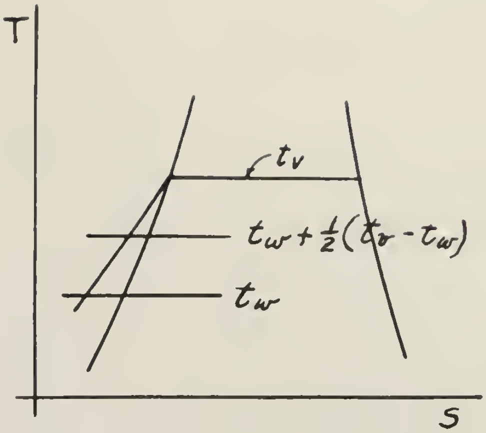
$$w = \frac{\rho\gamma}{3\mu} \sin\psi \cdot \chi_o^3 \tag{8}$$

Differentiating equation (8) and recognizing that both ψ and χ_o are variables:

$$dw = \frac{\rho\gamma}{3\mu} \left[3\chi_o^2 \sin\psi \, d\chi_o + \chi_o^3 \cos\psi \, d\psi \right] \tag{9}$$

Up to this point only the conditions of equilibrium have been considered. On the following page considerations of heat transfer will be considered.





Above there is shown a control surface drawn around an element of liquid flowing down the tube wall. The control surface is bounded on one side by the tube wall and on the other by the liquid interface. Two arbitrary cross sections, numbered 1 and 2, form the boundaries of the control surface on the top and the bottom. For simplicity in the sketch the curvature of the wall and the interface are not shown, but the dimensions of the control surface are x_o and $r_o\, d\psi$. The temperature of the condensate at the interface is the same as the vapor saturation temperature, t_v. The wall temperature is t_w, and if the thermal conductivity of the liquid is a constant, k, then a linear temperature distribution exists between the wall and the interface. Condensate which has formed above section 1 flows at a mass rate of flow w through the control surface. It is joined by a small mass dw so that both flow out of the control surface at cross section 2.

The average temperature of the liquid at all cross sections is $t_w + \frac{1}{2}(t_v - t_w)$. The liquid is subcooled to a temperature half way between the wall temperature and the vapor temperature. Condensate which is formed gives up more heat than the heat of vaporization, h_{fg}. Define

$$h_{fg}' \triangleq h_{fg} + \frac{1}{2} c_{p_l} (t_v - t_w) \tag{10}$$

then the heat transfered to the wall is

$$dQ = h_{fg}'\, dw \tag{11}$$

Using a simple Fourier expression for conduction

$$dQ = \frac{k}{x_o} r_o\, d\psi\, (t_v - t_w) \tag{12}$$

Equating (11) and (12) and solving for dw,

$$dw = \frac{k\, r_o\, d\psi\, \Delta t}{x_o\, h_{fg}'} \tag{13}$$

For ease in algebraic manipulation define a new symbol, E

$$E \triangleq \frac{3\mu k r_o \Delta t}{h_{fg}' \gamma \rho} \tag{14}$$

Equate expressions for $d\omega$ given by equations (9) and (13)

$$E\, d\psi = 3\chi_o^3 \sin\psi\, d\chi_o + \chi_o^4 \cos\psi\, d\psi \tag{15}$$

Now define another symbol, S

$$S \triangleq \frac{\chi_o^4}{E} \tag{16}$$

Note that

$$dS = \frac{4}{E}\chi_o^3\, d\chi_o \tag{17}$$

Substituting from (16) and (17) into equation (15) gives

$$d\psi = \frac{3}{4}\sin\psi\, dS + S\cos\psi\, d\psi \tag{18}$$

$$\frac{dS}{d\psi} + \frac{4}{3}\cot\psi \cdot S = \frac{4}{3}\frac{1}{\sin\psi} \tag{19}$$

Equation (19) is a differential equation of the first order and first degree, the general form of such equations being given in books on mathematics as:

$$\frac{dy}{dx} + Py = Q$$

in which P and Q are functions of χ only. The equation may be made an <u>exact</u> differential, and hence amenable of solution, if all terms be multiplied by

$$e^{\int P\,dx}\, dx$$

In the case of Equation (19) note that

$$e^{\int P\,dx} \cancel{} = e^{\int \frac{4}{3}\cot\psi\, d\psi} = e^{\frac{4}{3}\log(\sin\psi)} = (\sin\psi)^{4/3}$$

The solution of Equation 19 may then be set down as:

$$S = \frac{4.}{3(\sin\psi)^{4/3}} \int_0^{\psi} (\sin\psi)^{1/3}\, d\psi \tag{20}$$

The integral in the right hand expression may not be solved analytically so that graphical integration becomes necessary. The tabular values of the function S for various values of ψ were obtained from reference and are given on the succeeding page.

ψ	S	$S^{1/4} = \frac{\chi_o}{E^{1/4}}$
0	1.000	1.000
5	.963	.991
10	.991	.998
20	1.013	1.003
40	1.097	1.023
60	1.247	1.057
80	1.513	1.104
90	1.714	1.144
100	1.985	1.187
120	2.905	1.306
140	5.081	1.501
160	13.317	1.910
180	∞	∞

Local values of the film coefficient, h, for a given angle ψ are:

$$h_\psi \triangleq \frac{k}{\chi_o} = \frac{k}{S^{1/4} E^{1/4}} \tag{21}$$

Mean values of the film coefficient are given by

$$h_m\Big]_{\psi_1}^{\psi_2} = \frac{k \int_{\psi_1}^{\psi_2} \frac{r_o}{\chi_o} d\psi}{r_o(\psi_2 - \psi_1)} \tag{22}$$

$$h_m\Big]_{\psi_1}^{\psi_2} = \frac{k}{E^{1/4}(\psi_2 - \psi_1)} \int_{\psi_1}^{\psi_2} \frac{d\psi}{S^{1/4}} \tag{23}$$

Graphical integration must be resorted to in the solution of these equations. Reference (   ) gives the following solutions between the limits indicated:

$$h_m\Big]_0^{90°} = 0.953 \left[\frac{h_{fg}' \gamma^2 k^3}{3\mu r_o \Delta t}\right]^{1/4} \tag{24}$$

$$h_m\Big]_{90°}^{180°} = 0.652 \left[\frac{h_{fg}' \gamma^2 k^3}{3\mu r_o \Delta t}\right]^{1/4} \tag{25}$$

$$h_m\Big]_0^{180°} = 0.803 \left[\frac{h_{fg}' \gamma^2 k^3}{3\mu r_o \Delta t}\right]^{1/4} \tag{26}$$

Obviously the analytical solution is not valid for angles approaching 180°. When vapor is condensing inside horizontal tubes, the bottom of the tube fills with liquid and the thickness of the interface on the bottom of the tube must be given by some other expression. Equation (23) is valuable, however, from the top of the tube down to a certain angle. These observers were unable to find an analytical solution to <u>that</u> problem during this thesis work.

ORIGINAL DATA

Since the actual testing of the apparatus consisted of only two actual runs and a dry run to test the thermocouples all of the original data will be included in this report.

FIRST RUN

Top of test section

Thermo-couple Junction	Milli-volts	°F	Milli-volts	°F	Milli-volts	°F
1	4.06	217.4	4.06	217.4	4.06	217.4
2	3.79	206.1	3.80	206.5	3.82	207.4
3	3.91	211.0	3.69	202.0	3.71	202.8
4	3.82	207.4	3.62	200.8	3.66	199.1
5	3.50	194.1	3.53	195.4	3.50	194.1
6	3.47	192.8	3.45	192.0	3.49	193.7
7	3.41	190.2	3.60	198.3	3.61	198.7
8	3.45	192.0	3.45	192.0	3.45	192.0
9	3.29	184.9	3.34	187.1	3.34	187.1
10	3.49	193.7	3.45	191.5	3.45	192.0
11	3.36	188.0	3.36	187.6	3.36	188.0
12	3.45	192.0	3.45	192.0	3.45	192.0
13	4.34	229.0	4.35	229.4	4.33	228.6
14	4.39	231.0	4.35	229.4	4.37	230.2

Weight of condensate 109.5 lbs/hr and 114.68 lbs/hr

Boiling point of water 229.61°F

Barometer 764.4 mm of Hg

Barometer correction -3.25 mm Hg

Manometer 12.2" of Hg

Manometer correction -1.36" Hg

Total pressure 40.85" of Hg or 20.65 psia.

FIRST RUN

Side of the Test Section

Thermo-couple Junction	Milli-volts	°F	Milli-volts	°F	Milli-volts	°F
1	3.95	212.8	3.95	212.8	3.96	213.2
2	3.75	204.5	3.75	204.5	3.74	204.1
3	3.64	200.0	3.65	200.4	3.66	200.8
4	3.625	199.4	3.63	199.6	3.62	199.2
5	3.485	193.5	3.47	192.8	3.49	193.7
6	3.40	189.8	3.40	189.8	3.41	190.2
7	3.37	188.5	3.39	189.3	3.50	194.1
8	3.37	188.5	3.38	188.9	3.38	188.9
9	3.25	183.9	3.25	183.1	3.30	185.4
10	3.35	187.6	3.35	187.6	3.36	188.0
11	3.34	187.1	3.33	186.7	3.31	185.8
12	3.45	192.0	3.43	191.1	3.44	191.5
13	4.34	229.0	4.35	229.4	4.35	229.4
14	4.35	229.4	4.35	229.4	4.35	229.4

Weight of condensate	111 lbs/hr	109.5 lbs/hr
Boiling point of water	229.61°F	
Barometer	764.4 mm of Hg	
Barometer correction	-3.25 mm Hg	
Manometer	12.2" of Hg	
Manometer correction	-1.36" of Hg	
Total pressure	40.85" of Hg or 20.65 psia.	

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FIRST RUN

Bottom of Test Section

Thermo-couple Junction	Milli-volts	°F	Milli-volts	°F	Milli-volts	°F
1	3.91	211.1	3.90	210.7	3.905	210.9
2	3.73	203.7	3.72	203.3	3.73	203.7
3	3.59	197.9	3.66	200.8	3.65	200.4
4	3.56	196.7	3.58	197.5	3.59	197.9
5	3.45	192.0	3.44	191.5	3.44	191.5
6	3.32	186.2	3.34	187.1	3.32	186.2
7	3.19	180.5	3.20	181.0	3.20	181.0
8	3.19	180.5	3.19	180.5	3.20	181.0
9	3.05	174.8	3.05	174.8	3.04	174.4
10	3.05	174.8	3.04	174.4	3.04	174.4
11	2.80	164.4	2.80	164.4	2.80	164.4
12	2.83	165.7	2.82	165.3	2.78	163.6
13	4.35	229.4	4.35	229.4	4.35	229.4
14	4.35	229.4	4.35	229.4	4.36	229.8

Weight of condensate	111.75 lbs/hr	111.0 lbs/hr
Boiling point of water	229.61°F	
Barometer	764.4 mm of Hg	
Barometer correction	-3.25 mm of Hg	
Manometer	12.2" of Hg	
Manometer correction	-1.36" of Hg	
Total pressure	40.85" of Hg or 20.65 psia.	

SECOND RUN

Top of Test Section

Thermo-couple Junction	Milli-volts	°F	Milli-volts	°F	Milli-volts	°F
1	3.71	202.9	3.66	200.4	3.68	201.6
2	3.50	194.1	3.50	194.1	3.51	194.5
3	3.40	189.8	3.43	191.1	3.40	189.8
4	3.09	176.5	3.06	175.3	3.09	176.5
5	3.25	183.2	3.31	185.8	3.28	184.5
6	3.19	180.5	3.20	181.0	3.17	179.7
7	3.28	184.5	3.34	187.1	3.34	187.1
8	3.45	192.0	3.44	191.5	3.50	194.1
9	3.06	175.3	3.15	178.9	3.10	176.9
10	1.88	-----	3.04	174.7	3.13	178.0
11	1.04	-----	1.06	-----	1.04	-----
12	0.94	-----	1.00	-----	0.95	-----
13	2.40	146.2	2.92	169.4	2.45	148.2
14	4.10	219.0	4.10	219.0	4.10	219.0

Weight of condensate	81 lbs/hr
Entrance Manometer	6.0" of Hg
Manometer correction	-1.36" of Hg
Barometer corrected	30.35" of Hg
Total pressure	34.99" of Hg
Exit Manometer	5.05" of Hg
Correction	-1.10" of Hg
Total Pressure	34.20" of Hg
Weight of cooling water	83.5 lbs/minute

Temperature of cooling water in 54°F out 68°F

SECOND RUN

Side of Test Section

Thermo-couple Junction	Milli-volts	°F	Milli-volts	°F	Milli-volts	°F
1	3.67	201.2	3.70	202.4	3.68	201.6
2	3.45	192.0	3.43	191.1	3.49	192.0
3	3.38	188.9	3.31	185.8	3.38	188.9
4	3.025	173.7	2.98	171.8	3.00	172.7
5	3.19	180.5	3.18	180.1	3.19	180.5
6	3.14	178.4	3.15	178.8	3.15	178.8
7	3.20	181.0	3.20	181.0	3.25	183.1
8	3.48	193.4	3.52	195.0	3.48	193.4
9	3.20	181.0	3.20	181.0	3.22	181.9
10	3.22	181.9	3.25	183.1	3.12	177.5
11	2.88	167.8	1.34	-----	2.61	160.7
12	1.17	-----	1.15	-----	1.87	-----
13	2.74	-----	2.65	------	2.50	-----
14	4.10	219.0	4.11	219.4	4.11	219.4

Weight of Condensate	91.13 lbs/hr	91.5 lbs/hr
Entrance Manometer	5.99" of Hg	
Manometer correction	-1.36" of Hg	
Barometer corrected	30.40" of Hg	
Total pressure	35.03" of Hg	
Exit Manometer	5.00" of Hg	
Manometer correction	-1.10" of Hg	
Total pressure	34.30" of Hg	
Weight of cooling water	82.0 lbs/minute	

Temperature of cooling water in 54°F out 68°F

SECOND RUN

Bottom of Test Section

Thermo-couple Junction	Milli-volts	°F	Milli-volts	°F	Milli-volts	°F
1	3.74	204.1	3.73	203.7	3.73	203.7
2	3.50	194.1	3.50	194.1	3.49	193.7
3	3.46	192.4	3.44	191.5	3.45	192.0
4	3.02	173.5	3.00	172.7	3.00	172.7
5	3.225	182.0	3.17	179.6	3.14	178.3
6	3.05	174.8	3.08	176.1	3.00	172.7
7	3.01	173.1	3.00	172.7	2.98	171.8
8	2.95	170.4	2.98	171.8	2.93	169.6
9	2.76	162.8	2.71	160.7	2.72	161.4
10	2.50	156.3	2.49	149.9	2.44	147.9
11	2.21	142.7	2.31	146.9	2.26	144.7
12	1.58	-----	1.74	-----	1.74	-----
13	3.00	172.7	2.83	165.8	2.80	164.4
14	4.13	220.2	4.10	219.0	4.11	219.4

Weight of condensate	93.19 lbs/hr	96 lbs/hr
Entrance Manometer	5.95" of Hg	
Manometer correction	-1.36" of Hg	
Barometer corrected	30.36" of Hg	
Total pressure	34.95" of Hg	
Exit Manometer	4.85" of Hg	
Manometer corrected	-1.10" of Hg	
Total pressure	34.11" of Hg	
Weight of cooling water	83.0 lbs/minute	
Temperature of cooling water	in 54°F	out 71°F

Thermocouple Calibration Run

Thermocouple Junction	Millivolts	°F
1	4.30	227.35
2	4.30	227.35
3	4.30	227.35
4	4.30	227.35
5	4.30	227.35
6	4.30	227.35
7	4.30	227.35
8	4.30	227.35
9	4.30	227.35
10	4.30	227.35
11	4.30	227.35
12	4.30	227.35
13	4.30	227.35
14	4.30	227.35

Corrected barometer 40.74" of Hg or 20.0 psia.

Saturation temperature 227.96°F

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CALCULATIONS

Vapor Velocity Computations

1. First run:

Pressure 20.65 psia. Temperature 229.4°F. Average weight of condensate 111 lbs/hr or 0.0308 lbs/sec. specific volume at conditions indicated 20.089-.65(.897) = 19.506 ft^3/ lb.

$$\text{Area of } 1\tfrac{1}{4}'' \text{ pipe} = \frac{\pi D^2}{4 \times 144} \text{ ft}^2$$

$$= \frac{\pi \times 1.25^2}{576}$$

$$= .00851 \text{ ft}^2$$

$$\text{Area of } \tfrac{1}{2}'' \text{ pipe} = 0.00331 \text{ ft}^2$$

$$\text{Vapor velocity in the } 1\tfrac{1}{4}'' \text{ pipe} = \frac{\text{specific volume} \times \text{weight}}{\text{area of pipe}}$$

$$= \frac{19.506 \times 0.0308}{0.00851}$$

$$= 70.7 \text{ ft/sec}$$

For the ½" pipe Vapor velocity = 182 ft/sec

2. Second run:

Pressure 17.15 psia. Temperature 219°F. specific volume 23.27 ft^3/lb. Average weight of condensate 93 lbs/hr or 0.0258 lbs/sec.

Vapor velocity in 1¼" pipe = 70.4 ft/sec.

Vapor velocity in ½" pipe = 181.5 ft/sec.

The specific volumes used above were those of saturated vapor.

Overall Heat Transfer Coefficient U Computations

In these computations the value of h_{fg} used was that of saturated vapor since the quality of the steam used was nearly 100% and the slight difference will not affect the answers to any marked degree.

$Q = UA\Delta t_m$ is the formula used to determine U

Q = weight of condensate x h_{fg}

A = area of pipe

Δt_m = mean temperature difference between the vapor and the tube wall

U = Overall heat transfer coefficient

For 20.65 psia. and 229.4°F

$Q = 111 \times 959.0 = 106449$ Btu/hr

$$A = 2 \times 3.1416 \times \frac{0.875}{2 \times 12} \times 6.00 = 1.375 \text{ ft}^2$$

$\Delta t_m = 229.4 - 192.8 = 36.6$°F

Substituting into the formula U is 2110 Btu/ft^2 - °F - hr. and the value for the steam side coefficient h is 2180.

For the 17.15 psia. and 219°F run

$h_{fg} = 965.5$

$\Delta t_m = 219.0 - 184.1 = 39.6$°F

weight of condensate = 93 lbs/hr. (average value)

$Q = 93 \times 965.5 = 89792$ Btu/hr.

From the above information U is 1780

and as a consequence h = 1840.

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2. Jakob, Max, Mechanical Engineering, Vol. 58, No. 11

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